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This study examines the feasibility of a hydrokinetic (turbohydraulic) closed-loop marine propulsion transmission system in general, and particularly in a large hydrofoil vessel. As it is a synthesis of existing knowledge and hardware, this thesis primarily serves to collect the information needed for the conceptual design of such a transmission, and to present a procedure by which an engineer can quickly determine whether a hydrokinetic transmission is likely to meet his requirements.

The sample design calculation was based on a 750 ton hydrofoil craft, and the resulting system was compared with other propulsion transmission options, including mechanical shafting, superconducting electric drive, and various water jet systems. As the craft chosen was optimized for a water jet system and was far from the optimal application of a hydrokinetic transmission, the latter appeared to be only barely competitive with the water jets, and inferior to the other alternatives.

The author concluded that the potential application of the hydrokinetic system would be limited, although such a system was found to be technically feasible with existing hardware, and open to considerable improvement with a relatively small developmental effort.

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SYMBOLS

M	Mass flow rate (lbm/sec)
Q	Volume Flow Rate ft^3/sec unless otherwise indicated)
H	Hydraulic Head in "feet" (ft-lbf/lbm)
V	Velocity of a Fluid, absolute, or relative to a hypothetically stationary ship (ft/sec)
U	Velocity of a solid component, such as the speed of a turbine or pump rotor (ft/sec)
θ	Angle of turning, as in an elbow
W'	Relative Velocity of a Fluid in a rotating machine (ft/sec)
z	Number of blades in a turbine; height elsewhere
Re	Reynolds Number, Vx/ν non-dimensional.
D	Diameter (ft)
f	Friction factor
L	Length (ft)
x	A linear measurement of significance in a particular situation
ω	Angular velocity (rad/sec)
q	Heat transfer rate (Btu/hr or as labelled)
k	Heat Transfer Coefficient of a substance Btu/hr-ft-F^o
h	Heat Transfer Coefficient of a surface Btu/hr-ft^2-F^o
c_p	Specific Heat Btu/lbm-F^o
μ	Viscosity lbm/hr-ft
ν	Kinematic Viscosity ft^2/hr
Pr	Prandtl Number (non-dim.)
ρ	Density lbm/ft^3
T	Temperature, F^o
N	Revolutions per minute

N_s	Specific Speed. British-American Non-Dimensional Variety unless followed by a notation. See Conversion Table.
W	Weight (lbm unless otherwise labelled)
R	Reduction Ratio
G	Mechanical gear reduction ratio
n	Number (as of stages)
C, K	Coefficient (non-dimensional)
η	Efficiency (non-dimensional)
Δ	Change (across a stage)

SUBSCRIPTS

g	gear
p	pump
t	hydraulic turbine
e	"engine"; prime mover; gas turbine, etc.
x	heat exchanger
d	based on Diameter
d_r	drag
L	based on length, such as Reynolds Number
D	Dry, as in pump dry weight W_{pD}
l	line
H	hydraulic
b	bulk
m	mixed mean
st	stage
r	rotor
w	weight
i	inlet or inside
θ	tangential
M	meridional; usually ≈ axial in an axial turbine
sc	"screw" (propeller)
s	"specific" (N_s = specific speed)
opt	"optimal"
j	designating an integer; as in a summation
o	outside

T	total
sr	strut
sy	spray
fs	Schoenherr
n	nacelle or pod

ABBREVIATIONS AND ACRONYMS

CRPP:	Controllable, Reversible Pitch Propeller
BHP:	Brake Horsepower, measured at prime mover output shaft.
PHP:	Pump Horsepower, measured at pump input shaft.
SHP:	Shaft Horsepower, measured at the shaft before the propeller.
PC:	Propulsive Coefficient.
TC:	Transmission Coefficient.
THP:	"Turbine Horsepower," used to identify a type of dimensional specific speed based on horsepower in lieu of volume flow.
BAND:	"British-American Non-Dimensional"; used to identify that system of specific speed which is based on revs per second and non-dimensionalized.
END:	"European Non-Dimensional"; used to identify that system of specific speed based on radians per second, divided by π, and non-dimensionalized.
GPM:	If capitalized, identifies a form of dimensional specific speed using RPM, flow in gallons per minute, and feet head.
CFS:	If capitalized, identifies a form of dimensional specific speed as above except using flow in cubic feet per second.
RPM:	Revolutions per minute.
gpm:	Gallons per minute of fluid flow.
cfs:	Cubic feet per second of fluid flow.
DBH:	Developmental Big Hydrofoil. A proposed U. S. Navy craft in the early design stages. No information on this craft is exact or final.
SWATH:	Small Waterplane Area, Twin Hull, previously called "modcat," or Low Waterplane Area Catamaran.
LWP:	Low Water Plane area.
SES:	Surface Effects Ship.
SFC	Specific Fuel Consumption, lbm fuel/BHP-hr .

INTRODUCTION

An increasing awareness among marine engineers of the capabilities of hydraulics, including high torque to inertia ratio, infinitely variable speed ratio, relatively constant power compared with speed variations, and the resulting "gearing" effect of higher torque at lower RPM, remote heat transfer, and smoothness of operation, has caused some to believe that ships might do well with hydraulic transmissions. Since efficiency, or propulsive coefficient, is a primary consideration in all marine design, for fuel space and weight reasons if not for reasons of cost, the lower efficiency of a hydraulic transmission would only be acceptable in special applications, either where direct shafting is unacceptable, and where weight or volume considerations make an electric drive unacceptable; or where an unusual maneuvering requirement exists and bulk or weight is again a consideration.

The latter situation may exist in a tugboat or icebreaker application. Here a hydrostatic (positive displacement hydraulic) device has already been applied successfully. (See Appendix One) The former occurs in a number of circumstances. Where the prime mover is a unidirectional turbine, such as an aircraft derivative gas turbine, some reversing device is needed. This application of a hydrokinetic (turbine-type hydraulic) device has also been applied successfully, at least in an engineering sense (see App. One) in the GTS AURIS, and in the Fottinger Converter of 1905-30, the forerunner of both the Pametrade system's torque converter in AURIS, and the automotive hydraulic transmission.

Another potential application of a hydrokinetic drive would be for remote power applications where low volume is important. Such a situation

exists in the hydrofoil craft, or in the SWATH (small waterplane area, twin hull) vessel. The purpose of this thesis is to investigate the feasibility of such a remote type hydrokinetic marine propulsion drive.

This study examined the general form of hydrokinetic propulsion transmission systems, their possible arrangements, and the various components that make up the system. Turbines were examined for design choice, specific speed, efficiency, size, and weight. Pumps were similarly examined, with the added considerations of cavitation, suction head, and the need for reduction gearing. The required gearbox was also studied, for weight considerations. Ducting components, including elbows, branched connections, and variously shaped pipes were studied for determination of drag, flow losses, weight, and heat transfer capability. Finally, the transmission as a whole was examined, and a design procedure outlined.

In order to demonstrate that the proposed design procedure was workable, and to determine whether a hydrokinetic system was competitive with other transmissions, a transmission was designed for a large hydrofoil craft, the Navy's proposed 750 ton DBH. The design was evaluated in terms of weight and efficiency expected, and compared with other transmissions already conceived for that vessel.

HYDROKINETIC TRANSMISSION SYSTEMS

2.1 General Information

A hydrokinetic transmission system is a means of transmitting power from a prime mover to a load, in this case a marine screw propeller, by the use of pumps, turbines, and a fluid, in a closed loop. Such a system involves at least two rotors in one casing, as in the simple fluid coupling, but may utilize any compatible number of pumps and turbines in series and/or parallel flow. Mechanical reduction gearing may or may not be used. Ducting systems may be simple or very complex, depending on the vessel configuration as well as on the number and arrangement of turbomachines. The design of the individual turbomachines also has a bearing on the ducting arrangement. A centrifugal or mixed-flow pump may require one less elbow than an axial-flow machine for example.

If a vessel using a hydrokinetic transmission has two or more prime movers, or propellers, or both, the arrangement may be even more varied. Systems may be cross-connected mechanically or hydraulically, and may be arranged fore and aft or athwartships. The mechanical gearing, if used, may be offset or epicyclic, and this will also affect the ducting arrangement.

In theory, the simplest system would have a single prime mover, a single propeller, a single pump of high specific speed, a single turbine of low specific speed, and a minimum of connecting ducts, perhaps three elbows and two straight pipes. The use of a high specific speed pump and low specific speed turbine, with no mechanical reduction, seemed appealing, and was examined. The use of reduction gears, with various numbers of pumps, pump stages, and turbine stages was also investigated. System

selection was made on the basis of total weight, including fuel, for a given vessel size, speed, and range. The design procedure utilized is detailed in Appendix 2.

2.2 Turbine Design

With the prime mover and propeller considered known, the reduction ratio is set, and turbine design may be considered. Since reduction is generally substantial, a turbine of low specific speed is desirable. The lowest known specific speed water turbine is the Pelton Wheel. However, this design was not considered, due to its very narrow range of acceptable efficiency, and because it is a dry wheel type turbine. That is, with water as a working fluid, most of the rotor is actually spinning in air, at greatly reduced drag. This would be difficult to achieve in an underwater pod in varying flow conditions.

Among the commonly used water turbines, the Francis turbine is the next lowest in specific speed. Figure Four (28) shows the range of high efficiency to be 0.93-0.94 at Ns (THP) of 30 to 80 or at Ns (CFS) of 75 to 130. The equivalent Ns (BAND) ranges from 0.10 to 0.18, with 0.10 selected as a target for high reduction ratios, and 0.14 for maneuvering or variable speed range operation.

Although not generally used for incompressible fluids, the 90° inflow radial turbine (commonly used in Diesel turbosuperchargers) is also considered feasible. (28) For large ranges of efficient maneuvering, the Francis type would be selected, but either is satisfactory in the vicinity of Ns = 0.10, the best point for achieving reduction. The design point efficiency of the Francis turbine is expected to be about one percent higher, at 0.94, than the 90° turbine, and the range through which this efficiency

is maintained is larger, but the 90° IFR machine, with an assumed efficiency of 0.93, is likely to be substantially smaller in diameter.

Turbine shape is based on specific speed, and turbine size may be determined when specific speed, horsepower, and RPM are known. Head and flow rate may thus be determined, and passage suze calculated. For Francis turbines, the method developed by Bovet (21) was used; while for 90° turbines, the procedure described by Carmichael (36) was followed, with some simplification in the case of incompressible flow.

When size is known, turbine weight may be approximated by modeling the burbine as a centrifugal pump and using the equations presented by Percival (16). The required relations and procedures are described in Appendix 3.

2.3 Pump Design

At first appearance, it would seem that the combination of a low-speed turbine and a single high-speed pump, probably an axial-flow (propeller) type, would best accomplish the desired speed reduction. However, a pump of the required specific speed may or may not be feasible for a given system. Since much information on pump design remains proprietary, the feasibility of such a machine is often subject to doubt, as it was in the sample case. A reasonable number of parallel pumps of sufficient stages would certainly provide the required combination of head, flow rate, and specific speed, in any probable application. Since gearing is often needed to combine a number of pumps and a single prime mover, the effect of utilizing this gearing to accomplish all or part of the reduction was also investigated.

It became apparent that pump loading can be increased greatly if

cavitation is not a primary consideration. The proposed closed hydrokinetic system has a real advantage over the water jet here, for pump inlet pressure, and therefore net positive suction head (NPSH) can be set at any reasonable desired level. An optimistic pump efficiency, 0.90, was assumed for initial cycle calculations for the sample vessel.

According to an old and well-known chart by Moody, (29) Figure 5, large volume pumps may reach an efficiency of 0.92 at a specific speed (GPM) of 2500. Efficiency remains above 0.90 in the specific speed range of 1500 to 4500; corresponding to 0.087 to 0.262, peaking at 0.146, in BAND units. Thus the highest natural (one stage turbine, one pump of one stage) reduction ratio is from 1.46 : 1 to 2.62 : 1, depending on the desired efficiency. Additional turbine stages are likely to reduce efficiency rapidly due to draft tube, scroll and nozzle losses, and due to additional bends and their turning losses. An efficiency reduction of 0.015 per stage added is assumed. Pumps, however, are axial in general, as high specific speeds are used, and added stages are assumed to reduce efficiency only 0.005, each. Parallel pump arrangements are so dependent on geometry that no single figure can be assumed. This is reflected in added line losses, and in the weight of the required gearing.

Two possible approaches to pump design were evident. The designer could attempt to design a new pump fitted to the demands of this new type of system; with high blade loading, low weight, and high inlet pressure and NPSH to control cavitation, while keeping the entire pump heavily loaded. If these so far untried concepts were used, model tests would be required. Alternatively, only pumps within the spectrum of existing ones could be considered, for a feasibility study of a hydrokinetic transmission

system that could be built without any advances in the theory of pump design. The latter course was chosen. Therefore, certain conventional design rules were used. Head per pump stage could not exceed 500 feet. Tip speed of pump impellers could not exceed 250 ft/sec. Figure 9 (35) was used to size the rotor, after the stage counts of pump and turbine and the number of parallel pumps were chosen from specific speed and reduction ratio considerations.

With pump size known, Percival's (16) methods were used to determine pump weight. Since many of the pumps examined in the sample required mechanical reduction between prime mover and pump, the weights of both offset and epicyclic gears were needed. Percival's equations were utilized here, also. These procedures are detailed in Appendix 4.

2.4 Ducting Design

Methods of calculating ducting section dry weights, and fluid weights contained in duct sections, and of estimating the probable head losses in the ducts were found and used to compare various arrangements and combinations of turbomachines. The rules used are collected in Appendix 5. These equations, together with geometric limitations, were used to evaluate the "line efficiency" between pump and turbine, a quantity dependent on both vessel and system configuration. A few general statements may be made, however.

If geometry permits, there is an optimal elbow curvature at which the sum of bend (acceleration, separation) losses and length (frictional) losses is minimized. Gill, (15), found from literature that this occurs at $R/R_o = 4.3$, where R = radius of curvature of the bend at the duct centerline; and R_o = radius of circular pipe forming the elbow, or half the

hydraulic diameter of a rectangular section. In cases other than this one, splitters or turning vanes will be beneficial. The splitters should be so located that r_{ij}/r_{oj} = constant, for $j = 1$ to n flow channels, and the number of splitters is therefore $n - 1$. Gill showed that:

$$n_{opt} = 2.12 \text{ Ln} \frac{(R/R_o) + 1}{(R/R_o) - 1}$$

Losses can then be calculated for each flow path within an elbow, and averaged proportionally to that path's share of the total cross-sectional area. However, empirical values of loss coefficients are available for such optimized elbows, and these may be approximated by:

$$K_T = 0.2 \sin \theta$$

If two lines, supply and return, are restricted to a specified cross-sectional area, losses may be minimized by keeping the flow velocities equal in the two lines. If cross section is not important, larger lines have lower losses, but diffusers, and less so contractions, also have associated losses, so the total line length would determine the optimum arrangement. A limited cross-sectional area for lines was assumed in the sample for two reasons. First, the available volume is limited in most modern ships, due to either economic or habitability reasons even when it is not a critical design parameter. Strut thickness in a hydrofoil is a critical parameter, and comparisons with other systems would become less meaningful as the number of variables is increased.

In the hydrofoil sample, a number of geometries are worthy of consideration. The independent port and starboard systems would appear to have the lowest total line lengths and the most straightforward machinery

arrangements. However, this arrangement requires seven elbows per side in the typical configuration, and total losses might be lower in a transverse system with only five elbows per side. Depending on pump size as opposed to vessel's beam, either arrangement may require less straight piping, a factor which is actually minor in comparison to the so-called "minor losses" of bends and other fittings. No valve losses were considered. In practice, very "clean" valves should be used where the system ducting penetrates the hull, or just inboard of any separating device that may be required for retractable foil arrangements.

2.5 Drag, Heat Transfer, and External Cavitation

The effects of drag and heat transfer may support or oppose each other, depending on the configuration of the vessel and its machinery arrangement. A monohull or bihull displacement vessel will have a relatively high skin friction drag, but substantial area available for skin cooling, or volume for conventional heat exchangers. A hydrofoil, on the other hand, has only the small strut area available for cooling, but the higher speed and higher drag condition increases the heat transfer rate. Increasing the area available could conceivably so greatly increase the power required, that more heat would have to be transferred per square foot than before.

In general, a conventional internal heat exchanger is best avoided in pumped power applications due to weight, volume, pressure losses, and either scoop drag or auxiliary power drain. In special cases, as with the Fottinger Converter which pre-heated the boiler feed water, which was its working fluid, the heat transfer may not be a problem at all, but in most modern high performance vessels, particularly small ones, pro-

pulsion heat losses are not recoverable. Shell cooling might be considered, if the arrangement favors this approach. Russian experiments with shell cooling for the emergency Diesel generators of icebreakers indicated that an overall heat transfer coefficient on the order of 100 Btu/hr ft^2 $^\circ$F could be achieved at zero speed. As icebreakers have thick hull plating, but as plate thickness will probably not be the dominant factor, this figure may be taken as a slightly conservative estimate. The effect of ship speed on the cooling rate would be substantial.

In the hydrofoil case, a more definitive calculation is needed, since heating of the strut is an indicator of potential cavitation difficulties. Fortunately, the hydrofoil craft has the advantage of high speed, and does not have to dissipate heat at an equilibrium rate during the low-efficiency, high power, low speed condition encountered during takeoff.

Although the water flows past the strut of the hydrofoil very rapidly and is not expected to reach a temperature anywhere near that of the wall, nevertheless, some heating of the water must occur, and this may be expected to have an adverse effect on the incidence of cavitation. In the common theoretical model, the vapor pressure of plain water is used as a basis. This varies considerably with temperature, and it would appear that heated strut walls would have a substantial effect. However, empirical evidence (26) suggests that entrained or dissolved air and salt particles raise the cavitation threshhold from the theoretical 0.25 psia for 59° water to an observed safe value of 2.5 psia. To reach this figure, the water would have to be heated to 134°F, which would require a very hot strut. Effects of entrained air would increase as dissolved air is forced out of solution, but the effects of entrained salt would decrease as the solubility

of solids would increase at higher temperatures.

No usable experimental results were found to solve this dilemma. The only temperature-varied experimentation was based on an inert gas dissolved in a liquid metal, with no dissolved solids. (38) Certainly this question must be answered, but it would be a part of the model testing that should precede the construction of any foil sections in the sample case of the hydrofoil craft. This question would also make an excellent thesis in itself. For the purposes of this system design study, the external cavitation problem is regarded as minor relative to the other important temperature-dependent effect, namely the internal pressure required to keep the heated working fluid in the liquid phase and prevent cavitation at the pump entrance, without requiring excessive wall thickness in the piping. In the sample case, this overpressure varied from less than ten to hundreds of psi.

SAMPLE SYSTEM DESIGN

3.1 General Information

With the design procedure outlined and the needed information collected, a sample design could be carried through. This was carried out for a number of reasons. A sample calculation can best demonstrate the procedure, and can bring to light any oversight or insufficiency in the preceding work. A sample result can be compared with other systems to determine the feasibility and potential competitive position of a hydrokinetic transmission.

3.2 The Sample Vessel

In order to compare the hydrokinetic transmission with its competitors, the vessel chosen must be suitable for a number of systems, and design data on such systems must be available. The United States Navy has a conceptual design of a craft that meets this requirement, the 750 ton "Developmental Big Hydrofoil" or DBH. Variations of the DBH propulsion system have been popular design concepts in various Navy activities. Like many other research and development projects, this craft has encountered repeated delays on funding; and there is no firm commitment for its construction in any of these, or any other, forms. Propulsion system designs considered for this vessel, all based on General Dlectric's LM-2500 marine gas turbines, have included mechanical right angle ("Z") drive, both independent and cross-connected, (41) water jet systems (15, 43, 49) and the superconducting electric drive. (22)

This relative wealth of equally unproven information makes this craft an ideal example for comparing a hydrokinetic system design with its

competitors. In retrospect, however, a fast, light, non-maneuvering vessel like a hydrofoil appears not to have been the optimal application of a hydrokinetic transmission system. Since it was a non-optimum choice, the potential problems were expected to be uncovered.

Because of the desire to compare the hydrokinetic version of the DBH with the alternative systems, the vessel was changed as little as possible. The vessel's configuration, including strut dimensions, flying draft, hull form, and basic drag datums were maintained, to be altered only as required for the size differences among the various propulsor pods. The prime movers were also held constant, using two General Electric LM-2500 marinized aircraft derivative gas turbines, as were the propellers on all systems having propellers, except for the superconducting system. Displacement, cruising speed, and endurance were also set as constants. Therefore the design goal was minimum weight of fuel and propulsion system, adjusted for the pod drag penalty, so as to maximize the vessel's payload capability.

Again, this procedure was not optimal for the hydrokinetic drive. The choice of cruising speed was severely limited to permit comparison of all the systems with the same prime movers. The superconducting system and the cross-connected mechanical system could be operated on a single LM-2500 at 37 knots, while one of the water jet systems would not operate at 50 knots with two such turbines. Forty-five knots was selected, although a lower speed would be more advantageous to the hydrokinetic system when compared with a water jet, as most readers will be inclined to do.

Another possible approach could have been to select a cruising speed,

and optimize a craft around a given engine. Gill (15) found this to be the best way to optimize the propulsion system of a hydrofoil craft with gas turbine prime movers, but it was not used here for two reasons. First, the data available for comparisons was for a given displacement, prime mover, speed, and endurance, and based on minimum weight. Secondly, the design procedure is not strictly limited to hydrofoils; and Gill's findings may not necessarily be as applicable to other types.

3.3 Designing the System

The vessel described above was taken to have a fixed total displacement, required range, and cruising speed. The prime movers having been set, a propeller was chosen in accordance with Figures 2 and 3. The former is an envelope of achievable propulsive coefficients at the design point for existing hydrofoil craft, rather than the performance map of a single system. The latter is a net efficiency curve of a particular supercavitating propeller form tested behind a pod, for varying sizes of propeller and pod. With propeller and prime mover characteristics known, the reduction ratio was fixed at 3.58. This reduction could be accomplished by hydraulic machines entirely, or partially by mechanical gear reduction.

A number of system arrangements were tried, including mechanical reduction up to 3.5 : 1, with appropriate numbers of turbine and pump stages, and of pumps in parallel flow. Three computer programs were used, in conjunction with Figures 6 and 9, to compare the different arrangements. The first program, "Reduction Ratios," was used to determine what combinations of hydraulic machines accomplished the required reductions at satisfactory levels of head and specific speed. This program was based on an assumed line efficiency of 0.95, and it also assumed no

parallel turbines, due to the geometric constraints of the pod. The results of this program were examined, and the systems with the lowest number of machine components conducive to good specific speed were selected for further evaluation, as such systems were expected to be most efficient. The results for the systems selected for further study are found in Table 3a, and those not selected are presented in Table 3e.

The chosen systems were then examined for lowest weight by using the pump design chart (Figure 9) and a second computer program, "Dry Weights of Machines." This program continued to assume a line efficiency of 0.95. The relations contained in this program are those presented in the appendices for turbine and pump sizing, weight of pumps and turbines, and weight of gears. 90° IFR turbines were used, and both epicyclic and offset gear weights were determined. These results are presented in Tables 3b and 3f.

The systems having satisfactory weights were next examined for line and liquid weights and line efficiencies, using the third program "Fluid Lines." This program used the equations presented in the appendices to determine the head losses in the ducting, and the weights of the duct components and the liquid contained in them. The selected systems examined at this point included one with a 3.0:1 epicyclic gear, one pump and one turbine; and four arrangements with no mechanical reduction. Of the latter, two used no gear at all, and two used an offset gear to achieve parallel pump positioning. One of each of these pairs was an independent (port or starboard only) system and the other was a transversely connected system.

The line efficiencies of the latter systems were so poor that the program results actually indicated a "negative line efficiency." Of course, this resulted from the large error of the initial assumed line efficiency.

These systems would not work at all without additional pump stages to make up the severe head losses in the lines. No further examination of ungeared systems was required.

Since it was determined early in this study that the ideal natural reduction of a hydraulic system was 1.4, it was deemed advisable to examine a system with that characteristic. Among those systems already tried in the first two programs, one having a mechanical reduction of 2.5 was close to the desired ratio. However, examination of this system in the lines program revealed that the first system tested, the one with 3 : 1 gearing and a single pump, was superior.

The selected system was further examined. First, the heat transfer rate was determined in order to check the assumed overpressure required to prevent internal cavitation due to high fluid temperature. This system was also examined with both Dural and CuNi lines. In addition to the basic weight saving resulting from the lower density of Dural, the superior conductivity of this material caused a substantial reduction in the required overpressure and a further saving by reduction in the required line thickness. In all these calculations, the skin of the lines was required to withstand the system pressure without the aid of stiffeners or other structure. As a result of the second and third iterations, the line weight dropped approximately ten thousand pounds, while the line efficiency increased only 0.003. The final results are presented in Tables 3c and 3d.

Finally, the sample propulsion system was checked for off-design performance to ensure that the craft would take off at a reasonable speed as designed. Data from Rabel (32) indicated that supercavitating propellers have net efficiencies of 0.46 to 0.50 at typical hydrofoil takeoff speeds of

26 to 30 knots. A conservative 0.45 was assumed. Takeoff drag was determined to be 153, 000 lb. with the pod drag penalty correction to Gill's drag values as datum. The required SHP was thus 11, 745 per shaft. To generate this power, 146 cfs of turbine flow and 781 feet turbine head are required; requiring the pump and turbine to operate at 90% of design flow and 77% of design head. From assumed performance maps, Figures 16a, b, and c, turbine efficiency was taken to be 0.91, and pump efficiency 0.87. Line efficiency should go up for reduced flow and down for reduced pump head, and was therefore considered to remain 0.91. The transmission coefficient is thus .72, and 15, 900 BHP must be produced by the gas turbine. This is a 42% margin at 100°F, or 60.4% at 60°F, an ample allowance for waves and other factors.

3.4 Summary of Results

This investigation revealed that line losses are the dominant form of losses in the sample system if the hydraulic machines are used to achieve large reduction ratios. Ducting losses at branches and elbows necessary to connect the additional pumps and the independent radial turbine stages are substantial. Line and liquid weights also increased. The primary advantage of the 3 : 1 mechanically-reduced system was that it had a vastly simplified ducting arrangement, with no branches, no sharp bends, and a minimum of piping to connect the two turbomachines. This factor offset the machinery weight saving of the 2.5 : 1 system which was theoretically closest to optimal, because this system required two pumps, and therefore suffered branch losses.

In addition to the added bend losses, the systems using no mechanical reductions suffered from the relative inefficiency of smaller pumps, and

the greater machinery weight of more smaller machines. These problems could be reduced if small pumps were designed for efficiency rather than for reliably moving liquids for auxiliary purposes, and if less overall reduction were required, either in a different application, or if a higher propeller speed were chosen.

The primary problem, line losses, would be less severe in many vessels with different configurations, and could be reduced by redesign of the branched duct components. For instance, experimental work might be done on the use of internal vanes in divergences to convert them into the equivalent of a zero-degree divergence and a vaned long-sweep elbow, as in Figure 17.

The general trend of the propeller and water jet efficiency envelopes, Figure 2, shows that a relatively slow craft may be more suitable for a remote hydrokinetic drive. In any case, at least eighteen percent of the input power is lost, and must be transferred out of the working fluid. The wetted surface of the strut of the sample vessel was found to be adequate for this purpose, even if all of the energy loss is in the form of heat rather than noise. The cavitation on the outside of the strut would increase somewhat as the strut is heated up to thirty degrees.

3.5 Description of the Resulting System

TURBINE

90° Inflow Radial	Design Flow 189.5 cfs
One stage	Design Head 884 ft.
N_s = 0.10 (BAND)	Design efficiency 0.93 (assumed)
Rotor Diameter 3.56'	Dry weight 3409 lb.
Case Diameter 5.8'	Liquid weight 0 lb. (submerged)

PUMP

Inducer first stage	Case diameter 5.5 ft.
Mixed flow second stage	Design flow 189.5 cfs.
N_s = 153.09 (CFS) per stage	Design head 960 ft.
Max. impeller diameter 3.4 ft.	Dry weight 816 lb.
Inlet diameter 1.37 ft.	Liquid weight 937 lb.

LINES

Seven components. Design stress level 6000 psi. Dural walls.

Number	Component	Bend Angle	Length or Radius	
1	Contraction	30°		8'
2	Semi-ellipse duct	0°	26.5'	
3	Elbow	30°		5'
4	Elbow	45°		4.5'
5	Contraction	90°		3.5'
6	Triangular duct	0°	28'	
7	Contraction	90°		3.5'

3.6 Comparison with Alternate Systems

The hydrokinetic transmission arrangement finally selected was found to be comparable in weight and efficiency to a water jet system at the chosen speed. As stated above, this was not an optimum speed for the hydrokinetic system, but was very close to the optimum speed for the water jet systems, for the chosen combination of vessel weight and prime movers. Mechanically connected systems remain so superior in propulsive coef-

ficient that they are superior at any reasonable range, assuming they can be produced with satisfactory reliability.

It is also admitted that if superconducting electric systems are perfected to actually achieve the efficiency, weight, and volume claims of their most optimistic proponents, the hydrokinetic system will not be competitive. The latter may, however, be designed to be more compatible with the dynamics of a gas turbine and a screw propeller, and may be safer, more shock-resistant, cheaper, and relatively maintenance-free. Where these considerations overcome the lower level of efficiency, the hydrokinetic system may be chosen. The results of the comparison are summarized in Table 4.

A NOTE ON SYSTEM DYNAMICS

A closed-loop fluid system is, in general, not inherently stable. Any disturbance may continue to increase in severity, leading to either an unsatisfactory stable point with severe cavitation, or to self-destruction by acoustic vibration. The precise dynamic behavior of the system depends on the characteristics of the individual machines, since flow may vary either way with RPM depending on the specific speed and other parameters.

In order to stabilize the loop, a pressure control is required. This usually takes the form of an accumulator, a fluid capacitance with a piston and spring, or with pressurized air or other gas above a reservoir of liquid. A fluid loop of the size designed in this study would need a reservoir in any case to serve as an expansion tank and source of make-up fluid because of unavoidable leaks in the way of shafting, and probably at the joints provided for foil retraction, if any. Therefore, a reservoir containing air under pressure, and with provision for adding make-up fluid under pressure, is recommended. Its size is dependent on the magnitude of the system's transients, and was not determined.

The design of the individual turbomachines so as to maximize their compatibility and minimize transients or instability was not considered in this study. It is however a first-order problem, and should be considered in a real design. The reader is referred to Delahanty, Paynter, Vaughan, Spannhake, and Donsky (44, 45, 46, 47, 48) which analyze either turbomachines or closed-loops containing them. Particularly, Delahanty (44) presents a pair of analog computer models for a closed loop test stand including a pair of pump-turbines with prime mover and load, and

two accumulators. One model is linear, while the other is more widely applicable to systems, but suitable only for very sophisticated analog computers.

CONCLUSIONS

Examination of the available pertinent information indicates that the hydrokinetic propulsion system is technically feasible. Such a system could be built, and would function. However, it would be preferable to all alternatives only in very special cases, if ever. The preliminary calculations suggested that greater relative efficiencies occur at lower speeds, and that lower power levels with proportionally higher heat transfer areas are preferable. Such a system could be selected by the marine engineer planning to power a vessel at speeds below forty knots where the volume or weight of a conventional electrical system is not acceptable, and the flexibility requirements of some portion of the drive line, or the degree of maneuverability required, make a mechanical linkage system unsatisfactory; and the added efficiency of a screw propeller over a waterjet outweighed the added developmental cost and the (small) decrease in reliability anticipated.

Other potential special applications might be those where there is a use for large quantities of heated water, or a large auxiliary power demand when the vessel is not moving, or preferably both; perhaps on an Arctic drilling barge. At present, there seems to be insufficient demand to warrant the cost of development, even though this cost is likely to be relatively low.

RECOMMENDATIONS

The following work is needed to reduce the number of assumptions and thereby increase the accuracy of this study.

1) More (published) research on pumps, including
   a) investigation of potential efficiencies of smaller pumps
   b) pumping for power rather than to move liquids
   c) pumping in non-cavitating environments with high loading parameters

2) Examination of heating on cavitation in sea water.

3) Development of data on loss-free fittings and piping, including
   a) converging connections
   b) diverging connections
   c) stop or cutoff valves
   d) connections needed for retractable hydrofoils

4) Examination of pump-turbine combinations, especially their dynamics in incompressible flow and their propulsion control systems.

5) Examination of weight-reducing designs for ducting, and the effects of such changes on losses. For example, grid stiffening would greatly reduce the required weight of long high-pressure ducts. The vertical stiffeners could be inside, and the horizontal ones outside in the sample case, both being therefore parallel to the flows enveloping them.

6) Examination and testing of typical 90° IFR turbines and various forms of pumps for off-design behavior in incompressible flow, particularly far from the design point, as an input to dynamic studies of closed pump turbine systems.

7) In systems where heat transfer area is limited, the use of additives, or fluids other than water, should be examined.

8) In the sample case, or for any hydrofoil craft, the use of a lifting body in lieu of a round pod should be investigated, as this would reduce the unmitigated loss associated with a large-diameter nacelle. Also, the choice of a thicker strut should be investigated, as losses due to spray drag and wavemaking resistance may well be offset by the reductions in both internal losses and dry weight of ducts nearer round in cross-sections.

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APPENDIX 1

A Survey of Hydraulic Marine Propulsion

Hydraulic power transmission has a long history of service at sea. This is because fluid systems have certain definite advantages over other systems. Hydraulic transmissions surpass mechanical ones in smoothness of operation, controllability, and slow-speed operation. They are more adaptable in terms of location, not requiring proximity or alignment, but less efficient. Compared to electrical systems, hydraulic transmissions have great compactness, high torque-to-inertia ratio, better low-speed torque, but have slower high-speed response (or poor high frequency stability), lower efficiency, and greater long-distance transmission losses.

Large gun mounts, anchor windlasses, constant-tension winches, and many other relatively low-speed deck machines are commonly hydraulically driven. These machines are generally hydrostatic; that is, utilizing positive displacement movers such as rams, piston or gear pumps, and similar motors. The most popular drive of this class, called the "Waterbury gear" or "A-end B-end" drive, has been used for main propulsion on boats and dredges up to 900 horsepower. (5) Hydrostatic devices operate at high pressure levels, often 5000 psi, and require lubricating fluids, usually oil. This is both messy and highly flammable in the event of a high-pressure leak; and the fireproof ideal fluid is still an engineer's dream.

Most hydrostatic propulsion drives were designed for workboats. For instance, the TMP Mk. II 12000 hydraulically shifted gearbox manufactured by Henry Meadows Ltd. of Great Britain provides 19 HP per 1000 RPM up to 5000 RPM, weighs 42 1/2 lb., measures 18 × 18 × 13 in., is controlled by a single lever, automatically brakes the propeller in neutral, and

provides 83% of ahead RPM astern. As it is positive displacement, it can offer zero slip. The pump provides oil to bearing lubrication as well, and the control valve operates a clutch throw piston splined to the shaft. The transmission is epicyclic, with the sun as input. Ahead output is from the ring, and backing power is taken from the rack. (4)

Another hydrostatic drive was designed for catamaran houseboats and other such craft. It provides two 100 HP motor-propeller assemblies similar to outdrives, but fixed to the bottoms of the hulls. A 210 HP gasoline engine can be located anywhere in the boat, and no rudder is needed. A steering actuator is used to control the craft by differential propeller speeds. The pump, which is fixed to the engine, weighs 165 lb., runs up to 4000 RPM, 5000 intermittent, and at 2000 psig. provides 290 ft-lb. of torque. The motors include a 2-1 reduction, and each provides 235 ft-lb. torque. This system has a transmission efficiency of 81% with typical line lengths. The motors run at 2000 RPM, and weigh 90 lb. each. Their casings are aluminum, and their surface areas provide all the cooling needed. Fluid flows at up to 30 ft/sec, and the full charge of fluid circulates 40 times a minute at full speed. (6)

Fairey Hydraulics Ltd. offers a very different system, a 40 HP system with a reservoir-immersed gear pump. It also operates at 2000 psi, with a 22 gpm flow rate, but shifting is by a three-hose system between pump and motor. (7) A family of axial-piston transmissions by Hydra-Power Inc. provides infinitely variable speeds at constant horsepower; with a choice of 7.5 to 75 HP. Both pump and motor have cam plates. This unit is an in-line, or single casing type, a feature that reduces applicability, but also reduces line losses, and particularly limits the portion of the system that has to withstand 2000 psi. This system has an efficiency

in excess of 85%. Reverse speeds are "limited," and a minimum service of 10000 hours is claimed. (10)

Perhaps the most interesting hydrostatic device is found on a Thames River (England) tug, LOA 45', beam 12'6", draft 5'6", which has a Thornycroft T-400 Diesel turning 1900 RPM driving a Dowty variable-delivery servo-operated hydraulic pump, supplying fluid at 1900 psi to a Staffo Mk. 7 radial-piston high-torque, low-speed motor, rated at 3000 ft-lb. torque, at 3000 psi and 300 RPM. Only 1900 psi and 230 RPM are used at full output to the 4 1/2 foot propeller, while up to 280 RPM may be used at the lower torque rating of cruising without a tow. At 220 RPM and 95 horsepower, a bollard pull of 1. 55 tons is available. An 8 knot cruising speed at 280 RPM, light load, no tow, has been achieved, and four barges can be hauled at 250 RPM. This is another design that locks the propeller in neutral. (8) This vessel is particularly interesting due to the alternate use made of its hydraulic power: the wheelhouse can be jacked up and down three feet to clear bridges.

The other type of device, the class that has been used on sea-going ships in the past, is called hydrokinetic, hydrodynamic, turbohydraulic, rotodynamic, or rotary variable displacement. Such devices use turbines for hydraulic motors, and centrifugal type pumps (in the broader meaning of the term). Transmission efficiencies in excess of 88% have been achieved. (13) No remote drives have yet been used, but long lines' losses and added nozzles, scrolls, elbows and other components would reduce the efficiency.

Assuming a hull is so formed that the prime mover can be mechanically connected to the propeller in a straight or slightly offset but still

parallel line, no system has matched the mechanical coupling at steady forward speeds. However, geared propulsion systems are difficult to back, and not very satisfactory for multiple prime movers on a single propeller. Also, mechanically geared vertical drives (Z-drives) have been found unsatisfactory in reliability.

When steam turbines began to supplant reciprocating steam engines, some speed reduction was needed between the prime mover and the propeller. Gear technology was then lagging behind the need, and two other systems were tried. Electric propulsion was generally considered superior, and was the only one resurrected during the gear shortage of the Second World War. However, a conventional electric system (not superconducting) is much heavier than a hydraulic system of the same power. This is because "any magnetic material saturates at an inconveniently low flux density."(2) That is, the torque per pound of iron core material is set. Besides causing the propulsion motor to be both large and heavy, this problem increases response time and reduces acceleration. In addition, the electric motor and generator must be cooled internally, usually by large air passages, again increasing bulk. Hydraulic systems are inherently lighter and smaller than electric systems, and can be cooled by a more conveniently located heat exchanger because the working fluid carries the heat away from pumps and motors.

For these reasons, in 1905, Dr. Hermann Fottinger developed a hydrokinetic converter. This device was first used in 1909 in a ship of 8000 SHP, and the first device had an 83% efficiency. In 1914, the Imperial German Navy installed twelve Fottinger Converters in ships of 35000 HP. Unfortunately, efficiency tended to drop as reduction ratios were increased, and as gears came of age, this device became less appealing. The losses

were the flow and shock losses of additional rotor stages required to achieve the higher reductions. (12) The best technical information on the Fottinger converter comes from articles (13, 14) on a plant designed for the SS Konigin Luise. A steam turbine operating at 1180 RPM supplied 25000 HP. A peak efficiency at the design point of 88.6% was achieved, with a very flat efficiency curve remaining above 85% in the entire cruising speed range. The design reduction was 5.3:1, with a ten percent speed variation reducing efficiency only one percent. The manufacturer claimed that an efficiency of over 90% could be achieved by redesign of blade edges, which were not optimized in the model tested. Some losses were recovered by using the boiler feed water as the hydraulic system working fluid, thus preheating it and raising the overall transmission efficiency to 90.76% even without optimizing the blading.

At that time, there was little interest in unusually maneuverable vessels such as icebreakers and tugs, so the hydraulic system was dropped when gears were improved. During the second World War, a shortage of gears caused the renaissance and improvement of electric propulsion, while the hydraulic systems industry was overtaxed making gun mount and turret drives and aircraft systems, all having priority over ship propulsion of an experimental nature. This development and experience margin made the AC drive more efficient, and the DC drive was found to be excellent for multiple engine and constant-maneuvering control operations, as on the Wind class icebreakers which were being produced in quantity at this time. The electrical system had the added advantage of applicability to ship's services such as electrical generation in tenders and in towing winches of fleet tugs. Equivalent central hydraulic auxiliary systems,

which could be so adapted, are now commonplace in submarines, but were not then available.

This was the end of the hydrokinetic ship drive until another new development in marine propulsion caused engineers to look again at the transmission problem. That development was the gas turbine, a high-speed non-reversible prime mover not suited to the sudden torque loading of a mechanical clutch, but also not suited to cold-starting under inertial loads as heavy as a marine double-reduction gear and propeller. Especially when turbines were combined with other engines such as Diesels in CODAG systems, the hydraulic clutch became attractive. (9) (see Fig. 1e) Another new consideration is the higher-speed supercharged Diesel. This engine has an undesirable (that is, not flat) torque vs. RPM curve. When resistance varies, as in towing, or the torque curves of two engines do not match, a variable transmission ratio would be highly desirable. Very low speeds, as for maneuvering services, are also a problem in these engines, which have a relatively high idle speed. A high-torque, low-speed output variable ratio transmission is desirable to avoid the cost and vulnerability of a CRPP. (9)

A simple hydraulic clutch or "fluid coupling" is smoother than a mechanical one, and lasts longer, but still has unsatisfactory torque characteristics. One such drive (Fig. 1a) has a centrifugal pump and a turbine in one casing, with no stator blades between. The torque depends on, and is in fact proportional to, the ratio of speeds between input and output shafts. Thus, there is no torque at all if there is no "slip," and the greatest torque is applied when the output shaft is stationary. This is smoother than a mechanical clutch, and does not wear out, but it is less efficient,

and still loads the engine in an undesirable manner.

A great improvement can be made by adding stator blades or nozzles to redirect the flow entering the pump. This device, patterned after a 1-stage Fottinger, and called a "torque converter," has excellent torque characteristics because propeller torque is still strongly and directly proportional to shaft speed ratio, while engine torque is inversely and weakly proportional. The result is a steplessly variable gear ratio from the ship's mechanical gear ratio, or even slightly less in a transient, to more than twice that. (9) (See Fig. 1b.)

This combined clutch and variable reduction gear was used on the 5500 SHP gas turbine ship AURIS built for Shell Tankers Ltd., and using the PAMETRADA gear system. (11) A conventional friction clutch was planned for steady forward speed for efficiency, but difficulties with this device caused it to be abandoned, and the ship operated satisfactorily on torque converters at all times. The system utilized a forward and a reverse torque converter, both permanently attached to pinion quill shafts. (Fig. 1f.) Maneuvers were performed simply by draining one converter and filling the other. (1) The astern converter is rated at 12000 HP at 6000 RPM, and 68% efficiency; but AURIS' turbine offered only 5500 HP at 3840 RPM. (11)

While converters can be designed to have a higher efficiency in a narrower band, the preferred design has been to have a flat efficiency curve of 80% from 0.6 to 1.4 of the N_{prop}/N_{engine} range, with the maximum at 85% at about 0.8. (9) (See Figs. 1b, 1c.)

These devices, and hydrokinetic devices in general, have "room for considerable development." (2) This paper is a step toward one such devel-

opment; speed reduction by selective design of high-speed turbopumps and low-speed turbines designed as remote units for specialized applications like hydrofoil and SES craft, or LWP and submerged-pontoon catamarans, where long mechanical drives are unsatisfactory, and the best present system, the water jet, is inefficient compared to the screw propeller. The hydraulic system, like the electric drive, can be applied to multiple prime movers; and the reverse is true in a hydrofoil, where two gas turbines may be needed for liftoff and one is sufficient for flight, hydraulically driving two supercavitating propellers in pods on the fully submerged foils. (3)

APPENDIX 2

DESIGN PROCEDURE

This section outlines the design procedure used in the sample calculations, and also assembles and combines the information and methods contained in the following appendices on individual components.

GIVEN INFORMATION

A propulsion transmission system is a device intended to connect a propeller to a prime mover in a vessel. The basic vessel configuration, the prime mover, and the propeller are therefore known. Since most ships are designed to maximize effective payload in proportion to displacement, the plant weight is considered to include fuel; so the endurance must also be stated.

From this point, there are two logical procedural choices. The designer may select a given displacement, speed, and drag (both takeoff and cruise speeds and drags in the case of a hydrofoil), and attempt to minimize plant weight, including fuel. Or, he may choose only the desired speed, and optimize the vessel around the capability of a specific prime mover. Gill (15) found this to be the optimum method for hydrofoil craft using gas turbines and water jets.

PROCEDURE

1. With the propeller, speed, and prime mover stated, and drag at that speed known, the propeller and engine RPMs are known, for an assumed transmission efficiency. Therefore, the reduction ratio is known. The possible choices of pump and turbine stage numbers, which must of course be integers, are then chosen, and tested for acceptable stage specific speeds. Part or all of the desired reduction may be done with gearing be-

fore the pump or after the hydraulic turbine, or all reduction may be inherent in the hydraulic system.

2. Choices are made among the possible combinations, with a view toward optimizing efficiency (fewest components yields lowest line losses if no components are overloaded) or plant weight or the combination that yields the lowest overall weight, as desired. A number of potential winners are retained for future comparisons when line losses, wet weights, and other significant differences are included.

3. With stage specific speeds known, stage efficiencies, and then machine efficiencies, may be estimated from the appropriate figures. (Figs. 4, 5)

4. The product of pump and turbine efficiencies should be maximized, with estimated weights taken into consideration in the choice.

5. With stage specific speeds known, and relating power to head and flow, the required fluid flow rate and turbine head are computed.

6. Assuming equal velocities in supply and return lines of equal sizes to minimize line losses within a given line volume, the pressure drop across the turbine can be determined.

7. The required NPSH, and hence pump inlet pressure, is determined from Fig. 6. By assuming line pressure drops, max. system pressure is determined.

8. Using Fig. 8, the size of the Francis turbine is found.

9. Using the design procedure of Appendix 3, the size of the 90° IFR turbine is found. The selection of turbine type depends on usage. In general, the 90° type will be smaller, and better for reversing, while the Francis will be slightly more efficient, and will have a wider range of high efficiency.

10. With the maximum pressure known and a line size and shape selected, appropriate formulae such as hoop stress are used to determine the thickness of ducts and casings. If the thickness is too great, thick wall formulae are required.

11. Sufficient information is now available to determine the dry weights of pumps, turbines, and if arrangement is known, gears, and lines.

12. With arrangement set, line losses may be computed.

13. Line efficiency is compared with the assumed value, and previous steps are recomputed if necessary.

14. With all efficiencies known, the required heat transfer capability at the design point is known. If another point is considered critical in terms of heat transfer, off-design calculations or assumptions are required.

15. Heat transfer methods are examined, and selection is made.

16. Weight, volume, losses, and wet weights of the heat exchanger, if any are used, are computed. If losses are sufficient, recompute previous steps.

17. With new figures, find wet weights of all components, and total system weight without fuel.

18. With engine RPM, transmission efficiency, propeller net efficiency, SFC, BHP, find fuel weight for desired endurance. (If the ship size optimizing procedure is being followed, there are repeated iterations on line size, drag, speed, and possibly even stage counts and numbers of pumps.)

19. Fuel weight plus wet weight plus dry component weights are summed to find the total plant weight for each of the competing systems, and the winner chosen. (For the ship size optimization, the ratio of displacement to this sum is maximized.)

20. Other critical points are checked. There must be a sufficient power mar-

gin at takeoff speed in a hydrofoil, for instance.

21. Determine maximum steady and intermittent speeds. Heat transfer may be the limiting factor here.

22. Examine the system for water hammer and other dynamic problems, and recompute casing thickness and weight if needed.

23. Examine reversing characteristics. In a 90° IFR, this may be accomplished by reversing the nozzles of the turbine. This is likely to increase the turbine diameter, perhaps up to 20%.

APPENDIX 3

Reduction Ratio and Turbine Design

The available reduction ratio in a hydrokinetic system is related to stage specific speed by combining the basic definition of specific speed with the head and flow per stage, assuming head and flow rate are distributed evenly over series and parallel stages respectively. In all sample calculations, the number of parallel turbine stages is set at unity.

$$\frac{N_e}{N_{sc}} = R = \frac{N_{s\,pst}}{N_{s\,tst}} \cdot \frac{n_{ts}^{3/4}}{n_{ps}^{3/4}} \cdot \frac{n_{pp}^{1/2}}{n_{tp}^{1/2}} \cdot G$$

Examination of available data (References 16, 21, 28, 29, 30; Figs. 4, 5) indicates that good stage specific speed ranges are:

TURBINES

N_s	Range of high η in		Peak η in		Reduction target (lowest good η)	
	Francis	90^o IFR	Francis	90^o IFR	Francis	90^o IFR
CFS	75-130	75-110	.94	.93		
THP	30-80	30-40	.94	.93		
BAND	0.1-?*	0.1-.14	.94	.93	.10	.10

*Different sources lead to a range from .16 to .29

PUMPS (large flow rates)

N_s	Range of high η (over .90)	Peak η	Reduction target (highest good η)
GPM	1500-4500	.92 at 2500	
BAND	.087-.262	.92 at .146	close below 0.21 (allowed to vary, as all stage counts are integers.)

For one single-stage pump and one single stage turbine, the natural reduction ratio is ideally 1.46 and probably cannot reach 2.5.

FRANCIS TURBINE DESIGN

Much of this section is taken from Bovet (21) and refers to Figure 8, parts a, b, c. Fig. 8a defines the shape of the runner (rotor) passage in terms of small letter parameters, which are really ratios, comparing to a critical dimension R_{2e}, because r_{2e} is set equal to unity. This critical dimension is:

$$R_{2e} = \left(\frac{Q}{\pi} \frac{1}{0.270\,\omega} \right)^{1/3}$$

By entering Figure 8b with N_s (END), the values of the needed ratios are found. The appearance of the passage and the limits of the blade edges are shown in Figure 8c.

In the range of interest, the rotor diameter may be approximated as:

$$D_r = 2\,(0.25 + r_{oe})\,R_{2e}$$

and the nozzle entrance diameter approximated as $1.3 \times D_r$ in the non-reversing machine, or $1.6 \times D_r$ in the reversing case. The total diameter of a turbine with a compactly designed scroll is then 1.6 or $1.9 \times D_r$, respectively.

The rotor length is found by:

$$L_r = (\ell_e + b_o)\,R_{2e}$$

but the total length is too dependent on arrangement, staging, and draft tube shape to be approximated by a constant factor, or a simple function.

The blade design of Francis turbines for compact machinery cannot follow the conventional methods described by Bovet for fixed hydroelectric machines. A procedure similar to that given below for 90° IFR turbines is recommended. The non-reversing Francis machine may have a more compact nozzle ring than the 90° machine, but in reversible machinery, the Francis is likely to have a larger nozzle ring. This could be avoided only by split or half blades or some similarly complex mechanical arrangement.

NINETY DEGREE INFLOW RADIAL TURBINE DESIGN

Much of this section is adapted from a compressible flow radial turbine design procedure presented by Carmichael (36).

The loading parameter, ξ, or the number of rotor blades, is first assumed, and the relations:

$$\xi = 1 - 2/z = V_{\theta 1}/U_1 \quad \text{and} \quad \xi U_1^2/g_o = H_t$$

are used to find U_1. Then, since:

$$60\ U_1/\pi\,\mathrm{RPM} = D_r$$

the rotor diameter can be found. Diameter is of course minimized as z increases, but the ratio of weight to diameter rises, and efficiency falls, as z becomes too large.

Total diameter is again 1.6 or 1.9 times D_r, depending on whether reversibility is desired. In the final calculation, the turbine's diameter must be found by actual design of the nozzle ring, and scroll.

After repeated design cycles have established the desired parameters,

turbine design is continued by finding:

$$V_{\theta 1} = g_o H/U_1 \quad \text{and} \quad V_{m1} \geq 0.75\ (2\pi U_1/z)$$

Entrance and exit velocity vector triangles are used (in conjunction with an assumed hub diameter) to shape the rotor blades. Flow channel shapes (which are a foreshortened view of blade shapes in a 90° IFR machine) will be similar to Bovet's (Fig. 8c) at the same specific speed, remembering to convert to END. Since

$$N_s(\text{BAND}) = \frac{\frac{1}{2\sqrt{\pi}} \frac{V_{m2}}{U_{2sh}} \frac{r_{2sh}}{r_1} \left[\left(\frac{r_{2sh}}{r_1}\right)^2 - \left(\frac{r_{2h}}{r_1}\right)^2 \right]^{1/2}}{\xi^{3/4}}$$

r_{2sh} may be calculated, and, with blade shape, serves as a check on the assumed hub radius. In a multi-stage machine, the required shaft diameter must also be considered.

Returning to the entrance station, the blade's axial dimension "b" is found, then the nozzle pitch and chord, and the nozzle ring diameter. The flow speed into the nozzles determines the required added diameter of the compact scroll.

The nozzle ring radius is the rotor radius plus the nozzle CHORD length plus clearances if reversibility is desired. This dimension will exceed the normal nozzle ring diameter, and may even exceed the scroll diameter that would be required in non-reversing service. See Fig. 8d.

Turbine stage weight is calculated by assuming a radial turbine to be similar to a centrifugal pump. The dry weight of the pump is given by Percival (16) as:

$$W_D = 3.32\ N_s\ (\text{CFS}) D_r^2$$

As turbine stages are substantially separate at low specific speeds, this weight is attributed to each stage, with the conical annular connecting duct being considered equivalent to the first stage scroll and either of these equivalent to the centrifugal pump's discharge casing. The final draft tube is regarded as part of the ducting, as it will be a diffusing elbow in most probable arrangements.

APPENDIX 4

PUMP SELECTION AND REDUCTION GEARING

PUMP DESIGN

A number of parameters are available to pump designers. Some of these are redundant, and not listed here. They are summarized by Percival (16).

Specific Speed = $N\sqrt{Q}/H^{3/4}$ is usually expressed dimensionally, with H in "feet" and Q in gpm or cfs. Specific speed will be labeled GPM or CFS, to show which is intended, and this is not meant to imply that those are the actual units of N_s, which come out revolutions -feet$^{3/2}$-lbm$^{3/4}$ per minute-second$^{1/2}$-ft$^{3/4}$-lbf$^{3/4}$ in the CFS case.

According to one old but popular chart (fig. 5) the efficiency envelope of the largest pumps peaks at 0.92 for N_s (GPM) = 2500, and is above 0.90 for N_s from 1500 to 4500. Specific speed also determines impeller (rotor) shape, as shown on the bottom of this figure.

Efficiency is related to flow, head, and input power by

$$\eta_p = \rho\, QH/550 \cdot PHP$$

The net positive suction head (NPSH) required to avoid cavitation has been found to be proportional to output head in accordance with a factor called the "Thoma Cavitation Coefficient." For conventional values of this coefficient, σ, fig. 6 shows the relation between H and NPSH. NPSH may be used instead of H in computing specific speed, and the result is "suction specific speed," a measure of a pump's ability to function in cavitation-prone conditions, and even with cavitation actually occurring.

The flow coefficient $\phi = V_m/U$, and the head coefficient, $\psi = Hg/U^2$,

are constant for a given pump's operating range. Combining fig. 6 and the head coefficient relation would seem to suggest that a pump can put out any head if it spins fast enough and has a high enough inlet pressure. Conventional pump designs, however, limit impeller tip speeds to 250 ft/sec or less. A more analytically sound, but less used, procedure is to determine the "diffusion factor"

$$DF = 1 - W'_2/W'_{1s} + \Delta V_\theta/2(c/s)W'_1$$

where c/s is called "solidity" and is slightly in excess of unity. DF measures "blade loading"; and experimental evidence indicates that losses increase rapidly at values greater than 0.6. DF is a function of radius, and is largest at the hub. The hardest-working pump found in the literature searched had a DF of 0.66 at the hub, 0.4 at the tip, with $r_h/r_t = 0.7$, and a very high NPSH, in excess of 100 feet.

Available data on acceptable levels of ϕ and ψ are so limited that a pump size and performance could not be determined with a satisfactory degree of assurance by these methods. Instead, fig. 9 (35) was used to find pump impeller diameter.

Added head per stage was therefore limited to 500 feet, to keep within the limits of the chart. Tip speed was checked, and always kept below 250 ft/sec. Single stage efficiency was determined as a function of Q and N_s by fig. 5, and reduced .005 per additional stage. Pump dry weight was calculated using (16):

$$W_D = (300.7 + 46.4\, n_{p\,s\,st})\, D_i^{2.3}$$

where D_i is the inlet diameter; found from $D_i \pm D_r(B/D)^{1/2}$, where B/D and D_r are determined from fig. 9.

Both for reasons of blade loading and of cavitation, pump RPM is severely limited. With turbine speed set by propeller speed, further speed reduction may be accomplished more efficiently by gearing than by excessive numbers of parallel pumps. In fact, for a single prime mover, a point could be reached where added pumps plus coupling gearing would outweigh the reduction gearing allowing fewer pumps.

REDUCTION BY MECHANICAL GEARING

Reduction gear design is summarized by Percival (16) and only partially repeated here.

An integral gearbox provided with a prime mover is usually the lightest possible solution. Lacking this, a planetary gear should be used as long as there is a single pump shaft per prime mover. Pumps may be arranged in series on a shaft (although parallel in terms of liquid flow) as long as the resulting line losses and liquid weights are preferable to the heavier offset gearing, and the vessel's configuration permits the arrangement. Gear efficiency is assumed to be 0.98, dropping one hundredth per additional output shaft.

The K factor of the gear is taken at 500, for a lightweight hardened and ground gear. Then:

$$W_g = (0.35\ 126{,}000\ BHP_e/K\ N_p)(\Sigma Fd^2/C)(1.3 + 0.4\ n_p)$$

where the second group is dependent on "m" factors; and a pair of equations to evaluate these factors is tabulated for each gear arrangement, as follows:

GEAR TYPE	m equation	$\Sigma Fd^2/C$ equation
offset	$2m_g^3 + m_g^2 = 1$	$\Sigma Fd^2/C = 1 + \frac{1}{m_g} + m_g + m_g^2$
offset with idler	$2m_g^3 + m_g^2 = m_o^2 + 1$	$\Sigma Fd^2/C = 1 + \frac{1}{m_g} + m_g + m_g^2 + \frac{m_o^2}{m_g} + m_o^2$
double reduction double branch	$2m_g^3 + \frac{2m_g^2}{\frac{m_o+1}{m_o}} = \frac{m_o^2+1}{2\left(\frac{m_o+1}{m_o}\right)}$	$\Sigma Fd^2/C = \frac{1}{2} + \frac{1}{2m_g} + 2m_g + m_g^2 + m_g^2/m_o + m_o^2/2m_g + m_o/2$
planetary	$2m_s^3 + m_s^2 = \frac{0.4(m_o-1)^2+1}{b}$	$\Sigma Fd^2/C = \frac{1}{b} + \frac{1}{bm_s} + m_s + m_s^2 + \frac{0.4(m_o-1)^2}{bm_s} + \frac{0.4(m_o-1)^2}{b}$

Definitions:

m_g = ratio of pinion to larger mating gear

m_o = overall ratio, G

m_s = pinion to sun gear ratio

b = number of planets (not less than 3)

APPENDIX 5

Ducting Losses and Weights

Bernoulli's equation may be used to describe conditions in a duct.

$$P_1 + \frac{1}{2} \rho V_1^2 + \rho g Z_1 = P_2 + \frac{1}{2} \rho V_2^2 + \rho g Z_2 + \text{losses}$$

If the duct is not tapered, and the fluid is incompressible, $V_1 = V_2$. If the system is a closed loop, the Z term may be neglected, as a loss on one side is offset by an equal gain on the other side. Then ΔP = losses, assuming P to be constant over the station cross section, and these losses may be expressed in terms of a head loss, since $\Delta P/\rho = H$.

$H_{line} = \Sigma h_{ij}$ where h's are calculated for each component of the ducting as follows.

STRAIGHT PIPE:

$$h_{12} = \frac{f L}{D_H} \frac{V^2}{2g}$$

where the friction factor f is found from the popular "Moody" diagram (17). This equation and those that follow assume that turbulent flow with a constant pressure across a given cross section is occurring. This is true only after fifty diameters from any disturbance and 100 diameters after 180° of elbows. (39) The effects of tangential swirl, as from a pump or turbine outlet, may also be significant. However, for the purpose of comparing arrangements in conceptual design, the resulting errors should be proportional between the arrangements being compared.

ELBOWS:

$$h_{12} = \frac{1}{2g_o} \left\{ V_1^2 + (2\lambda - 1) V_2^2 - 2\lambda V_1 V_2 \cos \alpha \right\}$$

where $\lambda = 0.6$ in a circular elbow or segment and 0.75 if the flow channel

is rectangular.

However, rather than risking errors by assuming a velocity distribution and parcelling losses among channels as described in the text (see Section 2.4) the elbow loss coefficient of an optimally divided elbow has been assumed to be $K_T = 0.2 \sin \theta$; where $L_{equiv} = K_T D_H / f$, or $h_{12} = K_T V_1^2 / 2g$. Since conventional piping design sets K_T from 0.6 for a long sweep elbow to 0.9 for a standard elbow, the savings resulting from the use of splitters are substantial.

BRANCHED CONNECTIONS:

In the event that a number of parallel pumps are used, two elbows of less than ninety degrees for each pump, and two branched connectors, must be used. (Other arrangements, generally with higher losses, might be used to place parallel flow pumps on series shafting to save gearing weight.) Friction losses are found by summing that of each branch:

$$h_T = \frac{f_o \ell_o}{g d_o} + \frac{f_1 \ell_1}{g d_1} + \ldots$$

and mixing losses for each flow path are found from:

$$h_{01} = \frac{1}{2g_o} \left(\lambda_1 V_0^2 + (2\lambda_1 - \lambda_2) V_1^2 - 2\lambda_2 V_0 V_1 \cos \alpha'\right) \text{ for diverging}$$

flows and:

$$h_{10} = \frac{1}{2g_o} \left(\lambda_3 V_1^2 + V_0^2 - 2V_0 \left(\frac{V_1 Q_1}{Q_o} \cos \quad ' + \frac{V_2 Q_2}{Q_o} \cos \gamma' \right) \right)$$

for converging flows. Here, station 0 is always the line containing the total flow, station 1 is the end of the path being evaluated, the angles used to enter Figures 14d, e, f are between the main pipe's extended axis and the particular branch, and the λ's and prime angles are obtained from the figures.

CHANGES IN SECTION AREA:

Losses due to changes in cross-sectional area of ducting may be estimated by $H_L = K(\Delta V)^2/2g$, where K is from Figure 14b, for expanding sections; and neglected for gently contracting sections. For abrupt contractions, $H = ((1/C_c) - 1)^2 V_2^2/2g$, where C_c is tabulated in Figure 14c. Losses in scrolls are assumed to be part of turbine efficiency, which is reduced 1.5% per additional stage due to the connecting ducts substituted for the scroll. Their weights are also considered similar. Shape changes not involving either area or direction changes are considered lossless, with the friction loss over their length evaluated at an "average" hydraulic diameter.

COMPONENT WEIGHTS:

$W_{Dry} = \rho P t \ell$, where W is dry weight of the section, ρ is the density of the metal (557 lbm/ft^3 for Constantin and CuNi), P is the section perimeter, t the thickness, and ℓ the length. For t in inches, other linear dimensions in feet, and design stress set at 6000 psi,

$$t \approx \frac{\text{pressure (psi)} \times \text{diameter (ft)}}{1000}$$

so long as t/D is less than 1/240 in the indicated units. If this criterion is not met,

$$t = (D_i/2)\left(((6000 + \text{Pressure})/(6000 - \text{press.}))^{1/2} - 1\right)(12)$$

which is the thick-wall cylinder formula, in the same units.

$W_{wet} = A\ell\rho_{liq}$, which, for water is equal to 62.4 $A\ell$, with A and ℓ in feet^2 and feet respectively, and W in lbm. These weight relations are used for each branch of a Y connection, and are applied to the average diameter or area as appropriate in an expanding or contracting section.

APPENDIX 6

Drag Considerations

Skin friction drag is generally related to velocity and planform area by the use of a drag coefficient, C_{dr}. That is:

$$\text{Drag} = C_{dr} \frac{A\rho V^2}{2}$$

For a strut or other foil section, $A = \ell c$, and:

$$C_{dr\,sr} = 2C_{fs}\,(1 + 2\,t/c + 60\,(t/c)^4)$$

where C_{fs} is the Schoenherr friction coefficient, evaluated from

$0.242/(C_{fs})^{1/2} = \log_{10}(Re \cdot C_{fs})$, and this Reynolds number is based on chord length.

For a surface-piercing foil section if any, and always for the struts, the spray drag coefficient is:

$$C_{dr\,sy} = 0.03\ t/c \qquad \text{for } V/(gL)^{1/2} \text{ over } 3.0.$$

For a nacelle or pod, the form drag coefficient is:

$$C_{dr\,n} = C_{fs}(1 + 1.5(D_m/L)^{3/2} + 7\,(Dm/L)^3)$$

However, nacelles are generally combined with struts and often with horizontal foils as well. This results in an added drag term called "parasitic" drag. Hoerner (20) states that in the case of the typical (6:1) aircraft engine nacelle the parasitic drag is equal to twice the drag of the area of foil covered by the pod. He also states the form drag of such a pod is likely to be larger than this value, but of the same order of magnitude.

APPENDIX 7

Heat Transfer Considerations

Surface heat transfer relations are similar to those for drag, in that they involve friction coefficients and utilize some of the same inputs, and both depend on a boundary layer effect.

$q = U A \Delta T$, with q in Btu/hr, U in Btu/hr ft^2 $^\circ F$, A in ft^2 and ΔT in $^\circ F$. U is the overall heat transfer coefficient

$$1/U = 1/h_{inside} + 1/h_{outside} + t/k$$

where h is the units of U, and t the wall thickness in feet, and k in Btu/hr ft $^\circ F$, constant for a particular material at a particular temperature.

For h_{inside}, the McAdams correlation intended for round tubes containing fully developed turbulent flow is often used:

$$hD/k_b = 0.023\ (GD/\mu_b)^{0.8}\ (\mu c_p/k)_b^{0.4}$$

where G/μ is equal to $Q/A\nu$ and the other new symbols represent material properties constant at a particular temperature, usually the "bulk" temperature, the average temperature of the fluid in the pipe. These properties are tabulated in Rohsenow (19). The same correlation is applied to other duct shapes by substituting the hydraulic diameter D_H for D. The correlation is applicable to cases where L/D is over 50, the Reynolds number based on diameter is over 2300 and under 10^7, and the Prandtl number is between 1/2 and 120; but it is often used at higher Reynolds numbers for want of an equally direct method based on experimental evidence in the higher-speed flows. The effect of shorter entrance length,

L/D under 50, is a much greater heat transfer rate, as shown in Figure 15a, which is for oils. The results for oil and water were found to correspond almost exactly by Harnett (40).

The heat transfer coefficient may be related to the friction factor by the Colburn analogy, for all turbulent flow:

$$h Pr^{2/3}/c_p G = f/2$$

Over flat plates, or the outsides of large tubes or ducts, where the boundary layer is at first laminar, then develops and becomes turbulent:

$$h L/k = 0.037 \; Pr^{1/3} \; (Re_l^{0.8} - 15,500)$$

This relation holds for Pr over 1/2; and in the all-turbulent case the Colburn analogy also applies. Figure 15a shows the effects of developing flows.

APPENDIX 8

Sample Calculation for Heat Transfer

The sample calculation for heat transfer through the strut duct walls utilized conventional heat transfer correlations from Rohsenow and Choi (19). The example value is that after the third Dural line weight iteration.

Outside flow velocity 45 knots

Heat transfer area over triangular return duct 88.2 ft^2

Heat transfer area over elliptical supply duct 39.1 ft^2

Outside surface, supply duct

$$\frac{h_{20} L}{k} = 0.037 \, Pr^{1/3} \left[Re_L^{0.8} - 15,500 \right]$$

$$h_{20} = 0.037 (7.9)^{1/3} \left[\left(\frac{45(1.689)\ 1.382\ 3600}{0.044} \right)^{0.8} - 15,500 \right] \frac{.34}{1.382}$$

$$h_{20} = \frac{kt \frac{ft}{sec\ kt}\ ft\ \frac{sec}{hr}}{ft^2/hr} \quad \frac{Btu}{hr\ ft^2\ {}^oF} = 6117$$

Outside surface, return duct

$$\frac{h_{10} L}{k} = 0.037 \, Pr^{1/3} \, Re^{0.8}$$

$$h_{10} = 4145 \frac{Btu}{hr\ ft^2\ {}^oF}$$

Duct walls

$$\text{(Dural)} \quad k = 63.8 \frac{Btu}{hr\ ft\ {}^oF}$$

$$h_w = \frac{k}{t(ft)} = \frac{Btu}{hr\ ft^2\ {}^oF}$$

Inside surface, all ducts

$$h_i = \frac{k_b}{D_H} 0.023 \left(\frac{\frac{Q}{A} D_H \rho}{\mu_b} \right)^{0.8} \left(\frac{\mu c_p}{k} \right)_b^{0.4}$$

$$h_i = \frac{.37}{D_H} 0.023 \left(\frac{\frac{Q}{3} (3600) D_H (62.4)}{.027} \right)^{0.8} \left(\frac{1.65 \times 1.0}{.37} \right)^{0.4}$$

$$h_i = \frac{Q^{0.8}}{D_H^{0.2}} (2220)$$

Supply duct

$$D_{H2} = 2.097$$

$$h_{i2} = \frac{189.5^{0.8}}{2.097^{0.2}} (2220)$$

$$h_{i2} = 127092 \frac{\text{Btu}}{\text{hr ft}^2 \ {}^\circ\text{F}}$$

Return duct

$$D_{H1} = 1.088$$

$$h_{i1} = \frac{189.5^{0.8}}{(1.088)^{0.2}} (2220)$$

$$h_{i1} = 144915 \frac{\text{Btu}}{\text{hr ft}^2 \ {}^\circ\text{F}}$$

Overall

$$\frac{1}{U_1} = \frac{1}{h_{01}} + \frac{1}{h_{i1}} + \frac{t_1}{k_1} = \frac{1}{2569} \text{ (Dural)}$$

$$\frac{1}{U_2} = \frac{1}{h_{02}} + \frac{1}{h_{i2}} + \frac{t_2}{k_2} = \frac{1}{1047} \text{ (Dural)}$$

$$q = (U_1 A_1 + U_2 A_2) \Delta T$$

$$\Delta T = \frac{q}{U_1 A_1 + U_2 A_2} = 44^\circ\text{F} \quad \text{for Dural}$$

APPENDIX 8

```
CC                              REDUCTION RATIOS
CC      THIS PROGRAM TESTS VARIOUS GEAR RATIOS, PUMP NUMBERS, PUMP AND TURBINE
CC      STAGES, FOR SPECIFIC SPEED OF 0.15 IN TURBINE STAGE, AND DETERMINES PUMP
CC       STAGE HEAD, FLOW,AND SPECIFIC SPEED.  STAGES ASSUMED IDENTICAL.  SUFFIC-
CC       IENT NPSH IS ASSUMED.  LINE EFFICIENCY ASSUMED 0.95.
      KI=5
      KO=7
      DO 6 N=2,7
      DO 6 M=1,5
      DO 6 L=1,3
      DO 6 K=1,5
      G=N
      X1=G/2.
      X2=M
      X3=L
      X4=K
      XNPST=(0.555*(X4/X2)**.75)/(X1*SQRT(X3))
      IF(XNPST.LT.0.109.OR.XNPST.GT.0.21)GO TO 6
      YNPST=XNPST*1715.
      XHT=((1507.26*X4**.75)/(X1*XNPST*SQRT(X3)))**.80
      QT=135057.2/XHT
      QPS=QT/X3
      XHPST=1.05*XHT/X4
      IF(XHPST.GT.500.0)GO TO 6
      NX2=X2
      NX3=X3
      NX4=X4
      WRITE(KO,103)X1,NX2,NX3,NX4,XNPST,YNPST,QPS,XHPST
  103 FORMAT(F10.1,3I10,F10.3,3F10.1)
      GO TO 6
    6 CONTINUE
      END
```

```
CC                                   DRY WEIGHTS OF MACHINES
CC      THIS PROGRAM DETERMINES PUMP, TURBINE, AND GEAR DRY WEIGHTS, BOTH OFFSET
CC      AND EPICYCLIC, AND PUMP AND TURBINE BASIC DIMENSIONS, USING INPUT DATA
CC      FROM RESULTS OF REDUCTION RATIO PROGRAM AND FIGURE 9.
        KI=5
        KO=7
      1 READ(KI,101)GPS,XGR,NTS,NPP,NPS,PIMD,BOVRD,XMS,PB
    101 FORMAT(6X,F10.3,F10.2,3I10,4F6.3)
        IF(GPS.LT.0.0) CALL EXIT
        PIO=PIMD*SQRT(BOVRD)
        XNGR=XGR
        XNPS=NPS
        XNTS=NTS
        XNPP=NPP
        Y=500.7+40.4*XNPS
        WP=XNPP*Y*PIO**2.3
        Q=GPS*XNPP
        DSQ=((108.335*SQRT(Q))**1.333)/0.9
        D=SQRT(DSQ)
        DIA=D*60./(350.*3.14159)
        WT=(209.55*DIA**2)*XNTS
        IF(XNGR.LT.1.3)GO TO 501
        WE=(0.35*124000./500.)*(210./34.)*XNGR*((1./PB)+1./(PB/XMS)+XMS+XM
       1S**2+(0.4*(XNGR-1.)**2/(PB*XMS))*(1.+XMS))
        GO TO 502
    501 WE=0.0
    502 WG=3276.7*XNGR*(0.9+0.4*XNPP)
        WD=WP+WG+WT
        WDE=WP+WT+WE
        WRITE(KO,202)GPS,XGR,NTS,NPP,NPS,PIO,WP,WT,WG,WD,D,WDE,WE
    202 FORMAT(2F6.2,3I5,F6.2,F6.0,F7.0,2F8.0,F8.2,2F8.0)
        GO TO 1
        END
```

```
CC                                  FLUID LINES
CC     THIS PROGRAM DETERMINES LINE LOSSES, WET AND DRY WEIGHTS OF LINES, CHECKS
CC     LINE EFFICIENCY WITH INITIAL ASSUMED 0.95, AND SUMS ALL TRANSMISSION
CC     SYSTEM WEIGHTS.  PROGRAM ALSO PRINTS WALL THICKNESS OF PARTS AT WHICH HEAT
CC     TRANSFER CALCULATIONS ARE REQUIRED.
       DIMENSION WARY(50), WLIQ(50), PLOSS(50), XLOSS(50)
       KI=5
       KO=7
  500 READ(KI,100)N,Q,XNPP,RHOM,XNPS,RHOL,PID,SHEAD,TPHD,AMACH,ELASE,GRA
     1V,SIGMA,PTEMP
  100 FORMAT(I4,F6.1,F3.0,F7.2,F4.0,F4.2,F4.2,F6.2,F10.2,F7.0,F6.2,F7.2,
     1F6.0,F4.0)
       IF(N.LT.1) CALL EXIT
   10 READ(KI,101)I,COFIT,D,OFFA,OFFB,OFFC,ALAT,ABRAT,RZERO,ELZ,ELA,ELB,
     1PRSUR,EF,FFING,FTWO,YLAM1,YLAM2,YLAM3,ZLAM1,ZLAM2,ZLAM3,THETA,ALP
     2P1,ALPP2,BETAP,GAMAP,PFRAC
  101 FORMAT(I2,F5.0,F6.3,5F5.3,4F6.2,F5.2,3F5.4/6F4.2,5F5.3,F10.3)
       IF(I.LT.1.OR.I.GT.N)GO TO 500
       R=D/2.
       AL=3.1416*D**2/4.
       PRESS=((TPHD+SHEAD)*14.7/34.)*PFRAC+PTEMP
        IF(COFIT.LT.1.5) GO TO 11
        IF(COFIT.LT.2.5) GO TO 12
        IF(COFIT.LT.3.5) GO TO 13
        IF(COFIT.LT.4.5) GO TO 14
        IF(COFIT.LT.5.5) GO TO 15
        IF(COFIT.LT.6.5) GO TO 16
        IF(COFIT.LT.7.5) GO TO 17
        IF(COFIT.LT.8.5) GO TO 18
        IF(COFIT.LT.9.5) GO TO 19
       GO TO 10
CC     CONSTANT AREA ELBOW, COFIT = 1
   11 CONTINUE
       CLOSS=0.2*SIN(THETA)
       XLOSS(I)=CLOSS*(OFFA*Q*4./(3.1416*D**2))**2/(2.*GRAV)
       T=PRESS*D/(SIGMA*2.)
```

```
      RAT=T/D
      IF(RAT.GT.0.05) GO TO 401
      GO TO 402
  401 T=(D/2.)*(SQRT((SIGMA+PRESS)/(SIGMA-PRESS))-1.)
  402 WLIQ(I)=RHOL*3.1416*D**2*PZERO*THETA*(1.-FRSUB)/4.
      WDRY(I)=RHOM*3.1416*D*T*PZERO*THETA*(1.*(1.-FRSUB)+((RHOM-RHOL)/RH
     1OM)*FRSUB)
      FLOSS(I)=0.0
      GO TO 10
CC    RECT DUCT CDFIT=2, ARAT IS SIDE/BASE, D IS HYD DIAM, LLZ IS LENGTH
   12 X=SQRT(2.*ABASL/ARAT)
      Y=ARAT*X
      P=X+2.*Y
      WLIQ(I)=ABASL*RHOL*FLZ*(1.-FRSUB)
      T=(Y/2.)*(SQRT((SIGMA+PRESS)/(SIGMA-PRESS))-1.)
      WDRY(I)=RHOM*P*T*FLZ*((1.-FRSUB)+((RHOM-RHOL)/RHOM)*FRSUB)
      XLOSS(I)=0.0
      FLOSS(I)=FF*ELZ*(Q*QFRA/ABASE)**2/(D*2.*GRAV)
      WRITE(KO,993)I,T
  993 FORMAT(10X,' PART NO.=',I2,' THICKNESS(FT)=',F5.3)
      GO TO 10
CC    ROUND DUCT CDFIT =3
   13 P=3.1416*D
      WLIQ(I)=(1.-FRSUB)*3.1416*D**2*RHOL*FLZ/4.
      T=(D/2.)*(SQRT((SIGMA+PRESS)/(SIGMA-PRESS))-1.)
      WDRY(I)=RHOM*P*T*ELZ*((1.-FRSUB)+((RHOM-RHOL)/RHOM)*FRSUB)
      XLOSS(I)=0.0
      FLOSS(I)=FF*ELZ*(QFRA*W*4./(3.1416*D**2))**2/(D*2.*GRAV)
      GO TO 10
CC    EXPANDER, BENT OR STRAIGHT, CDFIT=4 D IS MAX DIAM, ARAT IS AREA FRACTION
   14   XK=CF.3
      XLOSS(I)=(XK+0.2*SIN(THETA))*(Q*QFRA/(ARAT*AL)-Q*QFRA/AL)**2/(2.*G
     1RAV)
      T=(D/2.)*(SQRT((SIGMA+PRESS)/(SIGMA-PRESS))-1.)
      WDRY(I)=RHOM*T*3.1416*D*ELZ*(1.+SQRT(ARAT))*((1.-FRSUB)+((RHOM-RHO
     1L)/RHOM)*FRSUB)/2.
```

```
      XLIQ(I)=(1.-FRSUP)*RHOL*FLZ*0.667*AI
      FLOSS(I)=0.0
      GO TO 10
CC    CONTRACTION, SEAT IS OK, COEIT=5, D IS MAX DIAM, RZERO CURVATURE
   15 XLOSS(I)=0.2*SIN(THETA)
      T=(D/2.)*(SQRT((SIGMA+PRESS)/(SIGMA-PRESS))-1.)
      FLOSS(I)=FF*FLZ*(Q/(AL*(1.+ARAT/2.)))**2/((2.*D)/(1.+SQRT(ARAT)))*2
     1.*GRAV)
      XLIQ(I)=(1.-FRSUP)*RHOL*FLZ*0.667*3.1416*D**2/4.
      XDRY(I)=THRUD*3.1416*D*FLZ*(1.+SQRT(ARAT))*(1.-FRSUP+((RHOM-RHOL)/
     1RHOM)*FRSUB))
      GO TO 10
CC    SPLITTER COEIT=6 D IS DIAM WITH MOST FLOW WHICH IS QFRC
   16 YLOSS=YLAM1*(Q*QFRC/AL)**2+(2.*YLAM1-YLAM2)*(QFRA*Q/(ARAT*AL))**2-
     12.*YLAM2*(Q*QFRC/AL)*(QFRA*Q/(ARAT*AL))*COS(ALPP2)
      YLOSS=YLOSS/(2.*GRAV)
      ZLOSS=ZLAM1*(Q*QFRC/AL)**2+(2.*ZLAM1-ZLAM2)*(QFRB*Q/(ABRAT*AL))**2
     1-2.*ZLAM2*(Q*QFRC/AL)*(QFRB*Q/(ABRAT*AL))*COS(ALPP2)
      ZLOSS=ZLOSS/(2.*GRAV)
      XLOSS(I)-YLOSS+ZLOSS
      DONE=D*SQRT(ARAT)
      DTWO=D*SQRT(ABRAT)
      XLIQ(I)=RHOL*(FLZ*AL+FLA*ARAT*AL+FLB*ABRAT*AL)*(1.-FRSUP)
      T=(D/2.)*(SQRT((SIGMA+PRESS)/(SIGMA-PRESS))-1.)
      XDRY(I)=RHOM*3.1416*T*(FLZ*D+FLA*DONE+FLB*DTWO)*(1.-FRSUB+((RHOM-R
     1HOL)/RHOM)*FRSUP)
      FLOSS(I)=(1./GRAV)*(FF*FLZ/D+FFONE*FLA/DONE+FFTWO*FLB/DTWO)
      GO TO 10
CC    COLLECTOR COEIT =7 D IS DIAM WITH MOST FLOW WHICH IS QFRC
   17 YLOSS=YLAM3*(QFRA*Q/(ARAT*AL))**2+(Q*QFRC/AL)**2-2.*(Q*QFRC/AL)*(
     1(QFRA**2/(ARAT*AL))*COS(BETAP)+(QFRB**2/(ABRAT*AL))*COS(GAMAP))
      YLOSS=YLOSS/(2.*GRAV)
      ZLOSS=ZLAM3*(QFRB*Q/(ABRAT*AL))**2+(Q*QFRC/AL)**2-2.*(Q*QFRC/AL)*(
     1(QFRB**2/(ABRAT*AL))*COS(GAMAP)+(QFRA**2/(ARAT*AL))*COS(BETAP))
      ZLOSS=ZLOSS/(2.*GRAV)
      XLOSS(I)=YLOSS+ZLOSS
```

```
      DONE=D*SQRT(ARAT)
      DTWO=D*SQRT(ABRAT)
      WEIG(I)=RHOL*(ELZ*AL+ELA*ARAT*AL+ELB*ABRAT*AL)*(1.-FRSUB)
      T=(D/2.)*(SQRT((SIGMA+PRESS)/(SIGMA-PRESS))-1.)
      BUOY(I)=RHOM*3.1416*T*(ELZ*D+ELA*DONE+ELB*DTWO)*(1.-FRSUB+((RHOM-
     1RHOL)/RHOM)*FRSUB)
      FLOSS(I)=(1./GRAV)*(FF*ELZ/D+FFONE*ELA/DONE+FFTWO*ELB/DTWO)
      GO TO 13
CC    HULLET, CONE CONTRACTION COEFT=6  ABASE IS ANNULAR END AREA, D IS CLOSED
CC    END DIAMETER   ANY ANGLE THETA IS OK
   16 BBASE=ABRAT*ABASE
      XK=QFRT
      IF(ARAT.LE.0.0)GO TO 403
      GO TO 404
  403 XK=0.0
  404 XLOSS(I)=(XK+0.2*SIN(THETA))*(Q*QFRA/BBASE-Q*QFRA/AL)**2/(2.*GRAV)
      RONE=SQRT(RZERO**2-BBASE/3.1416)
      DHYDR=(RZERO**2-RONE**2)/(RONE+RZERO)+D/2.
      T=(RZERO-RONE)*(SQRT((SIGMA+PRESS)/(SIGMA-PRESS))-1.)
      WEIG(I)=(BBASE*(1.+ARAT)/2.)*(1.-FRSUB)*ELZ*RHOL
      BUOY(I)=3.1416*(0.667*RONE+0.667*RZERO+.333*D)*RHOM*ELZ*T*((1.-FRS
     1UB)+((RHOM-RHOL)/RHOM)*FRSUB)
      D=DHYDR
      FLOSS(I)=FF*ELZ*(Q*QFRA/(BBASE*(1.+ARAT/2.)))**2/(D*2.*GRAV)
      GO TO 13
CC    SEMI-ELLIPTICAL PIPE COEFT =3  D IS LENGTH OF FULL MINOR AXIS
CC    ARAT IS RATIO OF MAJOR TO MINOR SEMIAXIS
   19 XLOSS(I)=0.0
      A=D/2.
      B=A*ARAT
      DHYDR=3.1416*A*B/(A+3.1416*SQRT((A**2+B**2)/8.))
      D=DHYDR
      FLOSS(I)=FF*ELZ*(Q*QFRA/ABASE)**2/(D*2.*GRAV)
      P=2.*A+3.1416*SQRT((A**2+B**2)/2.)
      WEIG(I)=RHOL*(1.-FRSUB)*ELZ*ABASE
      D=3
```

```
      T=(D/2.)*(SQRT((SIGMA+PRESS)/(SIGMA-PRESS))-1.)
      WDRY(I)= [illegible]*(1.-F+SUP+((PHC[illegible]-[illegible])/[illegible])*[illegible]SUP)*T*P
      WRITE(KO,999)I,T
999   FORMAT(1X,' PART NO.=',I2,' THICKNESS(IT)=',F5.3)
      GO TO 10
500   SLOSS=0.0
      SDRY=0.0
      SLIQ=0.0
      DO 300 J=1,N
      SLOSS=SLOSS+FLOSS(J)+XLOSS(J)
      SDRY=SDRY+WDRY(J)
300   SLIQ=SLIQ+WLIQ(J)
      TRWT=SLIQ+SDRY
      TCL=1.-SLOSS/TPHD
      XPUMP=(3.79+0.12*XNFS)*PID**2.5
      WLP=XPUMP*XNPP*3.1416*PID**2/4.
      SYSWT=TRWT+MACH+WLP
      WRITE(KO,203)SLOSS,SDRY,SLIQ,TRWT,SYSWT,TCL
203   FORMAT(1X,[illegible].2,F14.0,3F15.0,F15.3)
      GO TO 600
      END
```

TABLE 1

MULTIPLYING FACTORS

to correct

different expressions of

SPECIFIC SPEED

TO → / FROM ↓	N_s (GPM)	N_s (CFS)	N_s (THP)	N_s (BAND)	N_s (END)
N_s (GPM)	1	.047	.0159 $(\times\sqrt{S})$	5.83×10^{-5}	1.23×10^{-4}
N_s (CFS)	21.2	1	0.337 $(\times\sqrt{S})$	1.236×10^{-3}	2.608×10^{-3}
N_s (THP)	62.89 $(/\sqrt{S})$	2.97 $(/\sqrt{S})$	1	.00366 $(/\sqrt{S})$	.00722 $(/\sqrt{S})$
N_s (BAND)	17,151.	811.3	272.7 $(\times\sqrt{S})$	1	2.11
N_s (END)	8128.4	384.5	129.2 $(\times\sqrt{S})$	0.473	1

where S = Specific Gravity, or $\dfrac{\text{lbm/ft}^3 \text{ actual}}{62.4 \text{ lbm/ft}^3}$

If you have the number in the left hand column, you multiply by the number in its row and the column of the desired form.

TABLE 2

VESSEL CHARACTERISTICS

750 TON DBH

Hull: Planing hull of the "Delta" type, the best design for Canard hydrofoil configurations.

Bwl	39'9"	D	24'6"
Bmax	45'4"	Full Load	773.2 tons (for mechanically driven, cross-connected version)

Foils: NACA 16-212 30 ft below keel, 12 ft below FWL.

Loading 1200 psf.

Canard design 33% forward, inverted T, flap controlled,

67% aft, inverted Pi, flap controlled,

all rotated about transverse axes for retraction

Fwd Span 46 ft.	Aft Span 88 ft.
Aspect Ratio 5	Aspect Ratio 8

Mechanically propelled version:
Foilborne propulsion:
2 LM 2500 gas turbines, 2 supercavitating propellers

Cross-connected Z-drive with skewed axis and 4:1 reduction.

After struts: thickness (max) 1.5 ft., chord 12 ft.

Six square feet of cross section assumed available for shafting or ducting.

TABLE 3.

The following tables, 3a. through 3f., are the results of the computer programs in Appendix 8.. Line 1 of tables 3a. through 3d. applies to a system having one turbine and one pump, and a 3:1 epicyclic gear. This system is independent; that is there is no connection between the port and starboard drive units. Line 2 of these tables describes the same system, except that the ducting is fabricated of Copper-Nickel instead of Dural. In fact, all of the systems with the exception of line 1 are CuNi, for equal comparison. Lines 1 and 2 show final (third) iteration results, while lines 2 through 8 show first unsatisfactory iterations of rejected systems, for comparison. System (line) 3 through 6 have no mechanical reduction, and require three pumps of five stages, and five turbine stages. In systems 3 and 4 the pumps are arranged on a single shaft. In systems 5 and 6 the pumps are placed in parallel with offset gearing. Systems 3 and 5 are independent, while 4 and 6 are cross-connected by ducting. Systems 7 and 8 use a 2.5:1 gear, and have two pumps of two stages, and one turbine. System 7 uses an epicyclic gear and places both pumps on one shaft, while system 8 uses a parallel arrangement and an offset gear. Unsatisfactory combinations of machines which were not good enough even to enter into the "Lines" program are presented in tables 3e. and 3f. for comparison.

TABLE 3a.

	LC GEAR RATIO	TURBINES	PUMPS	PUMP STAGES	NS(ND)	NS(GPM)	PUMP FLOW	ST.HEAD
1.								
2.	3.0	1	1	2	0.201	3442.1	189.5	438.5
3.								
4.								
5.	1.0	5	3	5	0.207	3545.0	24.0	460.6
6.								
7.								
8.	2.5	1	2	2	0.170	2920.7	94.7	438.5

NS(ND) IS SPECIFIC SPEED OF EACH PUMP STAGE IN BRITISH-AMERICAN NON-DIMENSIONAL UNITS.
NS(GPM) IS SPECIFIC SPEED OF EACH PUMP STAGE IN COMMON PUMP ENGINEERING UNITS, BASED ON FLOW IN GALLONS PER MINUTE.
PUMP FLOW IS THE FLOW RATE THROUGH EACH PUMP IN CFS.
ST.HEAD IS THE HEAD IN 'FT' ACROSS EACH PUMP STAGE FOR AN ASSUMED LINE EFFICIENCY OF 0.95.

TABLE 3b.

CC	DTR	GR	NTS	NP	NPS	DPIN	PWT	TWT	GWT(O)	DRWT(O)	TIPSPD	DRWT(E)	GWT(E)
1													
2	3.56	3.00	1	1	2	1.37	816.	3409.	12787.	17012.	176.97	9031.	4805.
3													
4	2.33	1.00	5	3	5	0.55	397.	3944.	6885.	16227.	123.19	9241.	0.
5													
6													
7													
8	3.33	2.50	1	2	2	0.97	742.	3408.	13934.	18085.	176.94	7165.	3014.

DTR IS DIAMETER OF TURBINE ROTOR FOR 90 IPS WITH NS=0.10
GR IS MECHANICAL GEAR REDUCTION RATIO BEFORE PUMP
NTS IS NUMBER OF TURBINE STAGES IN SERIES
NP IS NUMBER OF PUMPS IN PARALLEL
NPS IS NUMBER OF STAGES IN SERIES IN EACH PUMP
DPIN IS PUMP INLET DIAMETER
PWT AND TWT ARE PUMP AND TURBINE WEIGHTS IN POUNDS, PER MACHINE
GWT IS GEAR WEIGHT, (O) IF OFFSET, (E) IF EPICYCLIC
DRWT IS DRY WEIGHT OF ALL MACHINERY

TABLE 3c.

CC	LINE LOSS FT	DRY WT LB	LIQ WT LB	TRANS WT LB	SYS WT LB	LINE EFFIC
1.	76.36	1530.	9651.	11181.	21155.	0.916
2.	76.36	45403.	9651.	55054.	65028.	0.916
3.	1169.[illegible]	34883.	9331.	44213.	53555.	0.492
4.	1055.0	42345.	12604.	54950.	64291.	0.542
5.	1210.00	32426.	[illegible]	41261.	57503.	0.472
6.	1355.41	37331.	10182.	47573.	63801.	0.422
7.	1115.11	10124.	8013.	18136.	25304.	-0.272
8.	1240.75	10840.	9112.	19953.	38040.	-0.415

TRANS WT IS SUM OF WET AND DRY WEIGHTS OF ONE TRANSMISSION SYSTEM
SYS WT IS SUM OF TRANS WT AND MACHINERY WEIGHT AND MACHINERY
LIQUID WEIGHT, FOR ONE TRANSMISSION, NOT INCLUDING PRIME MOVER OR
ITS AUXILLIARIES, AND NOT INCLUDING PROPELLER, SHAFT, OR STRUCTURE
LINE EFFIC IS LINE EFFICIENCY, DEFINED AS LINE LOSS DIVIDED BY
TOTAL PUMP HEAD, AND ITS ACCURACY DEPENDS ON ITS PROXIMITY TO 0.95

TABLE 3d.

STRUT DUCT WALL THICKNESSES

1.	PART NO.= 2	THICKNESS(FT)=0.05[illegible]	PART NO.= 6	THICKNESS(FT)=0.009
2.	PART NO.= 2	THICKNESS(FT)=0.1[illegible]3	PART NO.= 6	THICKNESS(FT)=0.183
3.	PART NO.= 2	THICKNESS(FT)=0.058	PART NO.=18	THICKNESS(FT)=0.138
4.	PART NO.= 2	THICKNESS(FT)=0.058	PART NO.=18	THICKNESS(FT)=0.133
5.	PART NO.= 2	THICKNESS(FT)=0.058	PART NO.=18	THICKNESS(FT)=0.138
6.	PART NO.= 2	THICKNESS(FT)=0.058	PART NO.=18	THICKNESS(FT)=0.138
7.	PART NO.= 2	THICKNESS(FT)=0.058	PART NO.= 6	THICKNESS(FT)=0.043
8.	PART NO.= 2	THICKNESS(FT)=0.058	PART NO.= 6	THICKNESS(FT)=0.043

TABLE 3e.

CC GEAR RATIO	TURBINES	PUMPS	PUMP STAGES	NS(RD)	NS(GPM)	PUMP FLOW	ST.HEAD
1.5	3	2	4	0.209	3591.5	49.0	423.0
1.5	4	2	5	0.200	3421.0	41.2	402.9
1.5	5	2	5	0.169	2894.5	50.1	403.0
2.0	2	2	3	0.172	2942.4	62.5	445.0
2.0	5	1	5	0.179	3070.0	72.1	455.6
2.5	2	1	3	0.194	3328.9	125.0	443.0
3.5	1	1	2	0.172	2956.4	169.5	435.5

TABLE 3f.

CC DIR	GR	NTS	NP	NPS	LPIN	PWT	TWT	GWT(E)	DPWT(E)	TIPSPD	DRWT(E)	GWT(F)
2.00	1.50	5	2	4	0.67	388.	6591.	8501.	15540.	142.00	7980.	1001.
2.70	1.50	4	2	5	0.65	401.	7529.	6301.	16590.	154.00	9250.	1001.
2.58	1.50	5	2	5	0.64	382.	6961.	8361.	17705.	126.31	10343.	1001.
3.10	2.00	2	2	3	0.76	495.	5167.	11148.	16615.	154.00	7453.	1771.
2.58	2.00	5	1	5	0.95	448.	6952.	6525.	17725.	126.25	11171.	1771.
3.10	2.50	2	1	5	1.10	551.	5167.	10650.	15574.	154.00	6752.	3014.
3.50	3.50	1	1	2	1.37	815.	5409.	14916.	19145.	170.97	11429.	7234.

TABLE 4 Comparisons of Transmission Systems All Weights in Pounds

Item	Mech.	X-C Mech.	Su. Cond.	Water Jet Systems: Gill	Water Jet Systems: Kruse	Water Jet Systems: Study (49)	Hydro-kinetic	X-C HK
Sys. Wt.	97, 570	120, 020	128, 110	162, 009	223, 503	202, 000	109, 640	132, 090
TC	.95	.95	.930	—	—	—	.775	.768
PC	.551	.551	.540	.4229	.4984	.484	.450	.445
POD PENALTY	3.4%	3.4%	1.45%	1.0%	4.0%	3.1%	7.4%	7.4%
EHP	18, 000	18, 000	17, 600	17, 600	18, 100	17, 900	18, 700	18, 700
BHP	32, 700	32, 700	32, 600	41, 700	36, 400	37, 100	41, 600	42, 000
SFC	.442	.442	.440	.414	.426	.426	.414	.410
Fuel Wt.	649, 000	649, 000	648, 000	764, 000	694, 000	652, 000	771, 000	786, 000
Sys. + Fuel	746, 570	769, 020	776, 110	926, 009	927, 503	854, 000	880, 640	918, 090
Remark	Reliability subject to debate. Ready now.		Long lead time. High risk, untried.	High reliability. Ready now.			Presumably medium reliability. All components tested, although combination is untried.	

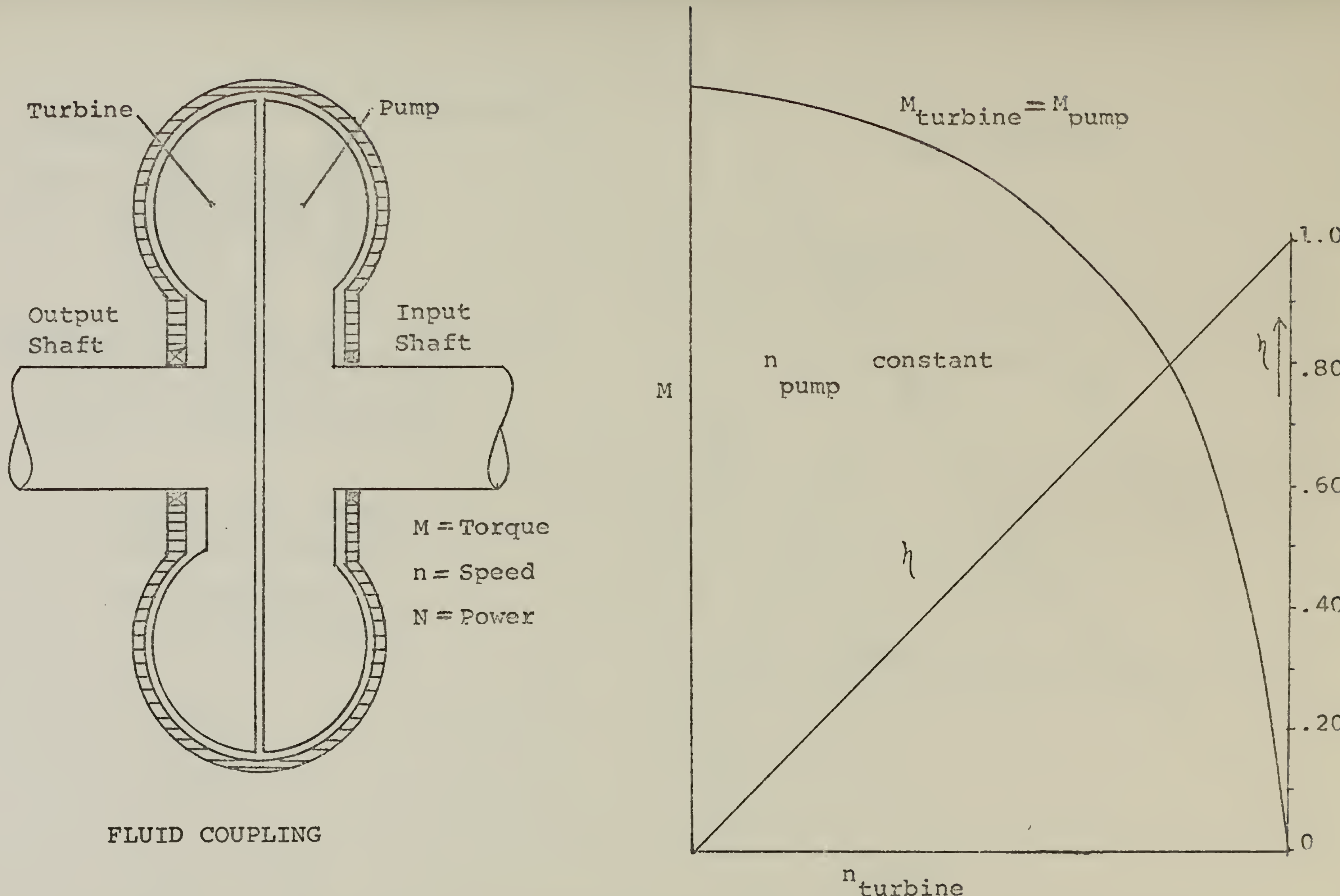
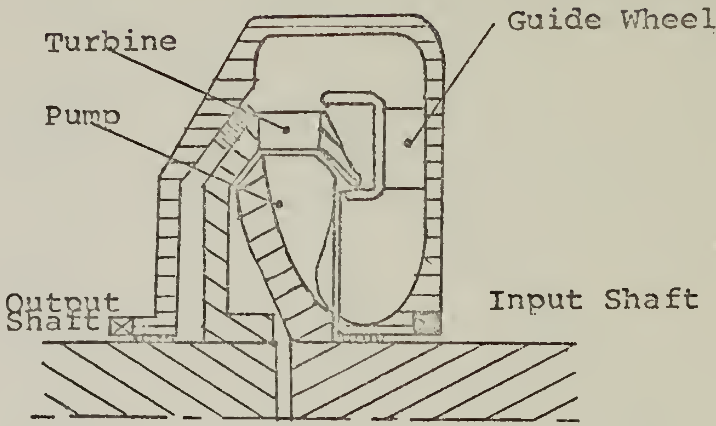
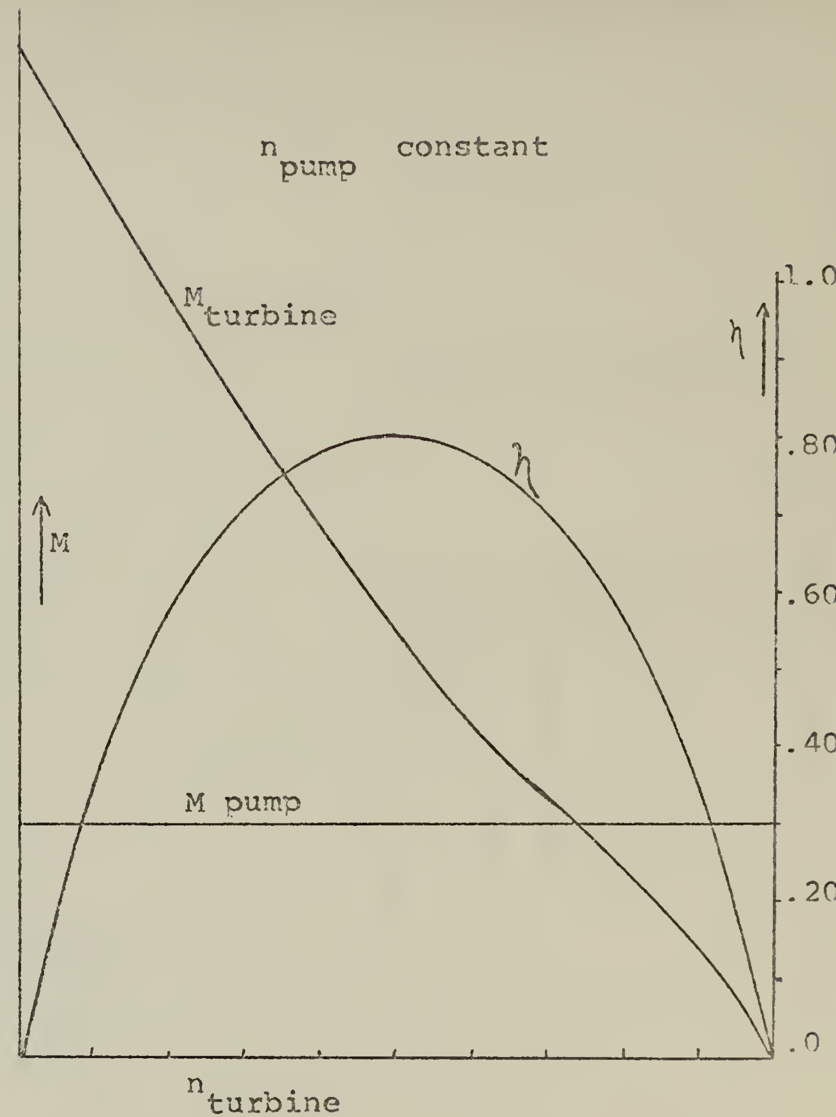


FIGURE 1a. FLUID COUPLING SCHEMATIC AND PERFORMANCE CHARACTERISTICS (9)





HYDRODYNAMIC TORQUE CONVERTER SCHEMATIC AND PERFORMANCE CHARACTERISTICS

M Torque

n Speed

N Power

FIGURE 1b. (9)

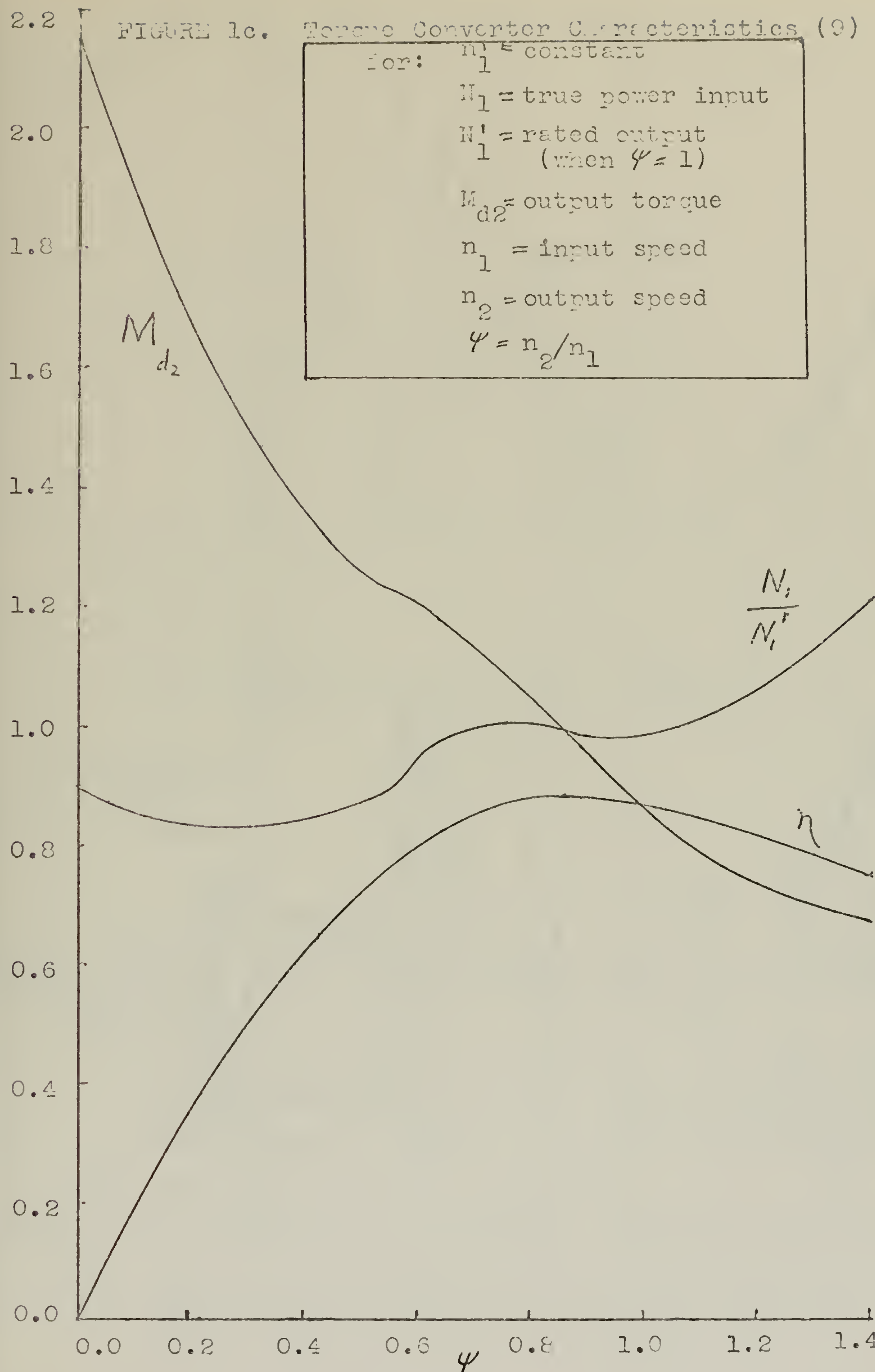


FIGURE 1c. Torque Converter Characteristics (9)

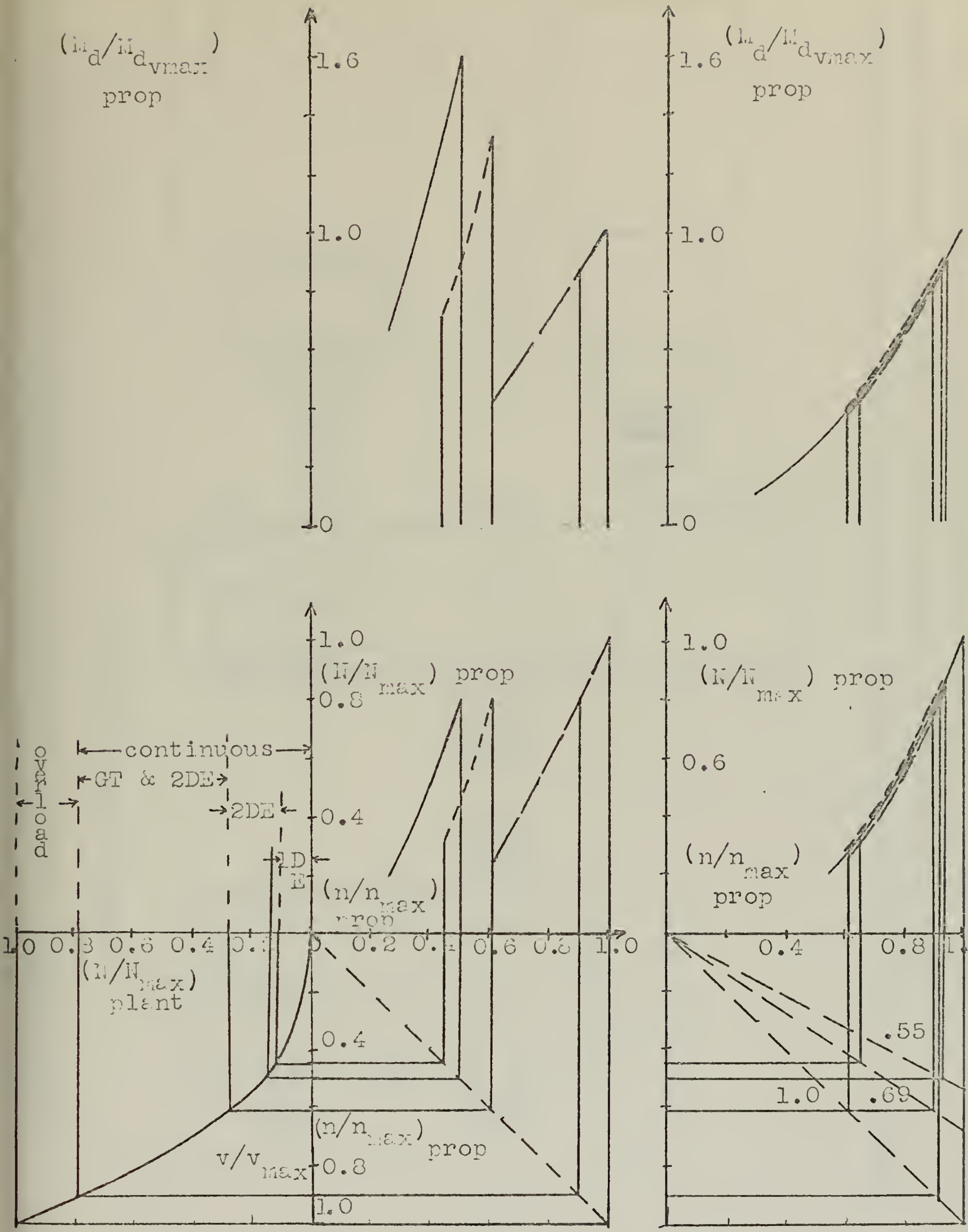


FIGURE 1d. ADVANTAGE OF VARIABLE GEAR RATIO IN OPTIMIZING ENGINE TORQUE MATCHING TO PROPELLER (9)

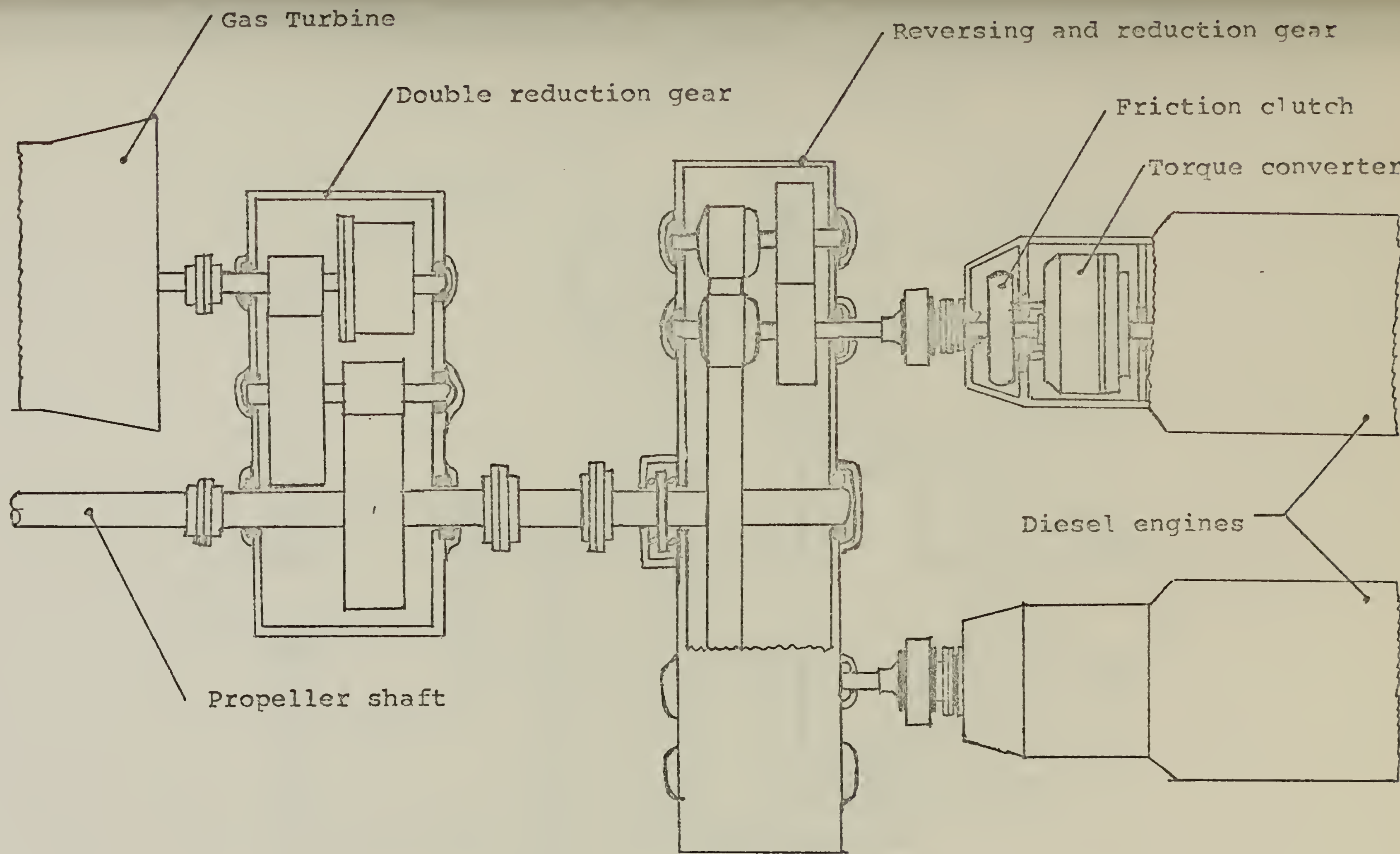


FIGURE 1e. CODAG SYSTEM USING TORQUE CONVERTERS ON DIESEL ENGINES FOR MANEUVERING AND WHEN GAS TURBINE AND DIESELS ARE ALL OPERATING. FRICTION CLUTCHES ARE UTILIZED FOR EFFICIENT LOW-SPEED CRUISING ON DIESELS ONLY. (9)

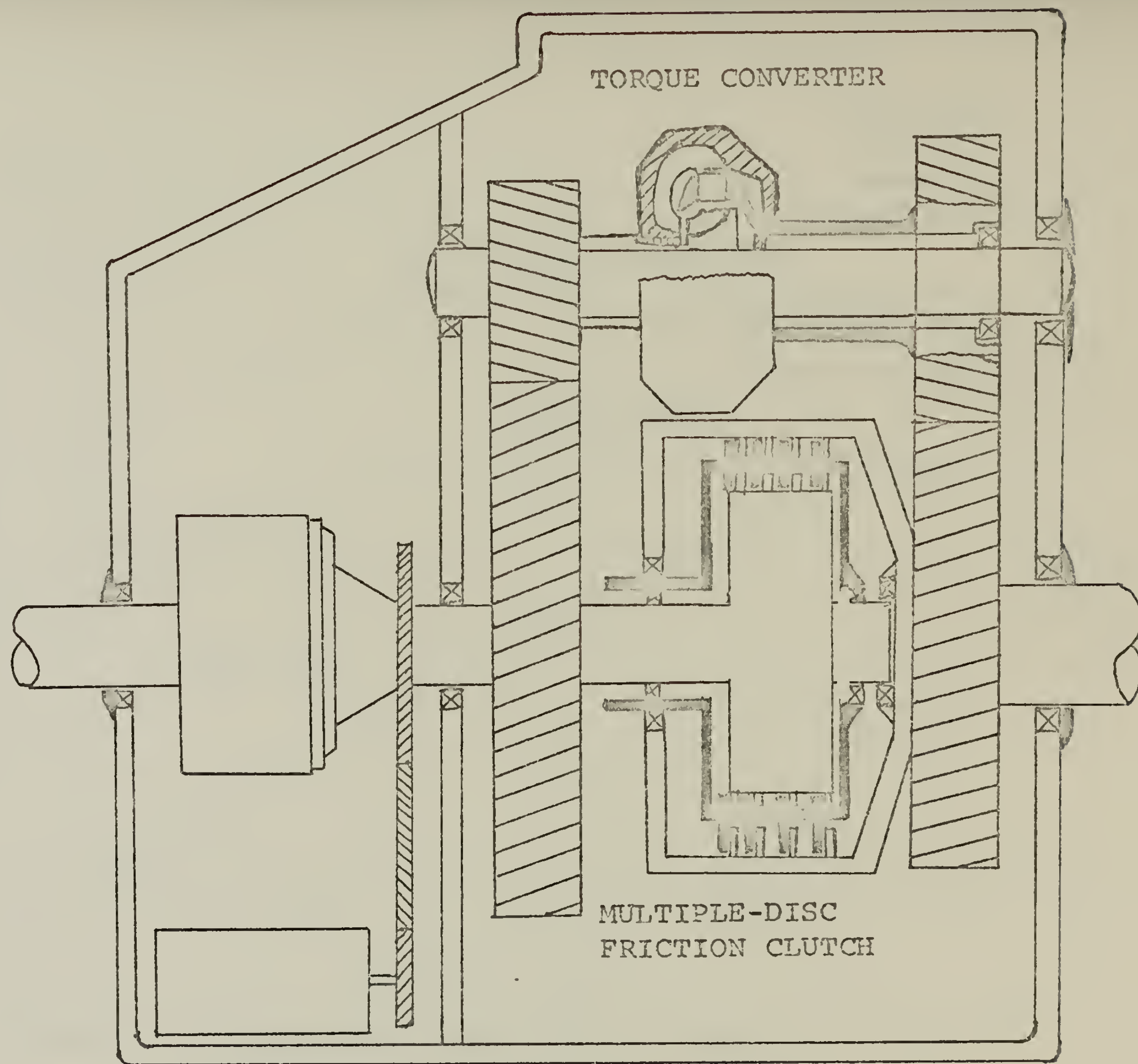


FIGURE 1f. REVERSING GEAR WITH TORQUE CONVERTER (9)

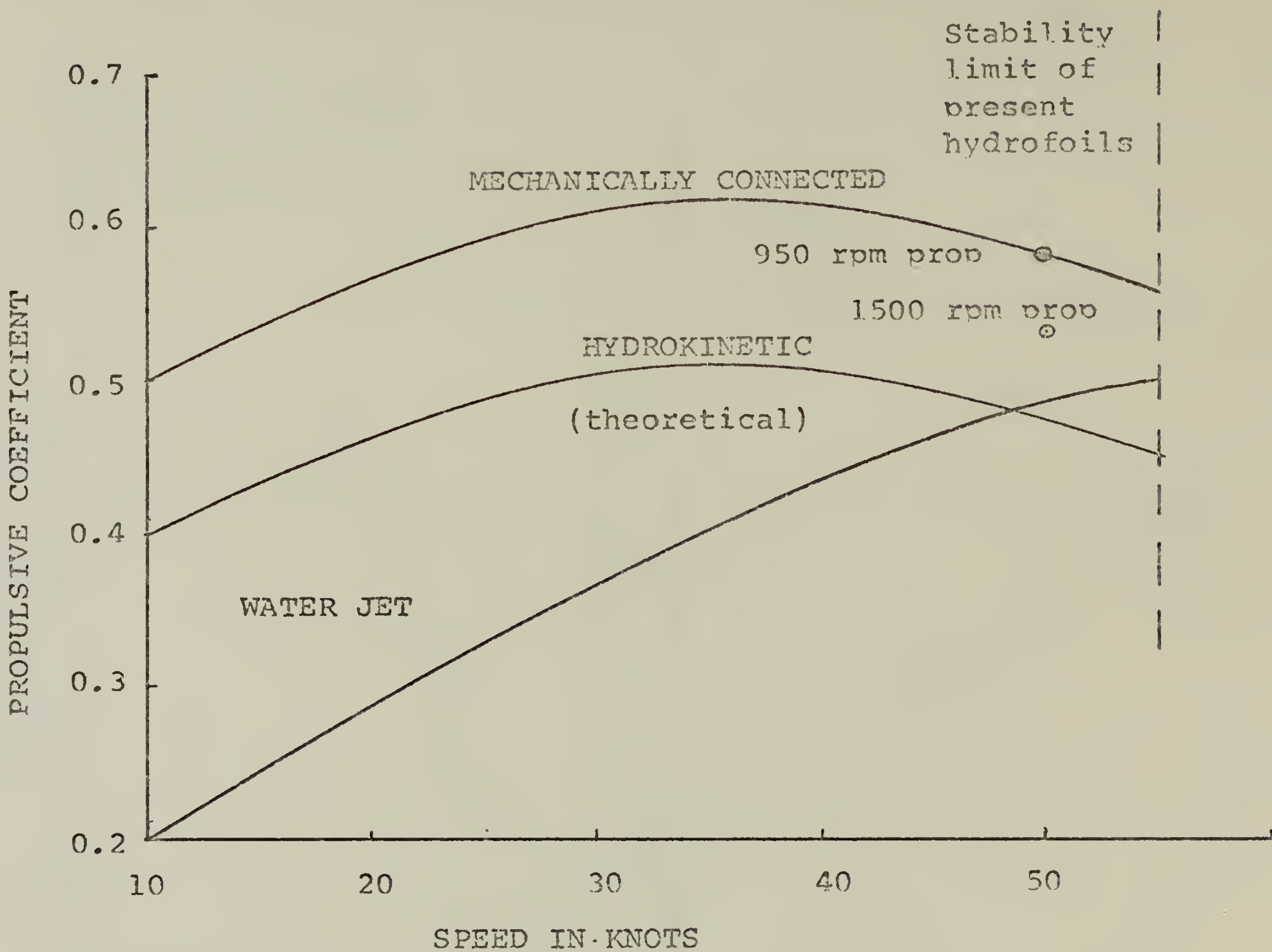


FIGURE 2. PERFORMANCE ENVELOPES OF HYDROFOIL PROPULSION SYSTEMS AND TRANSMISSION DEVICES (22)

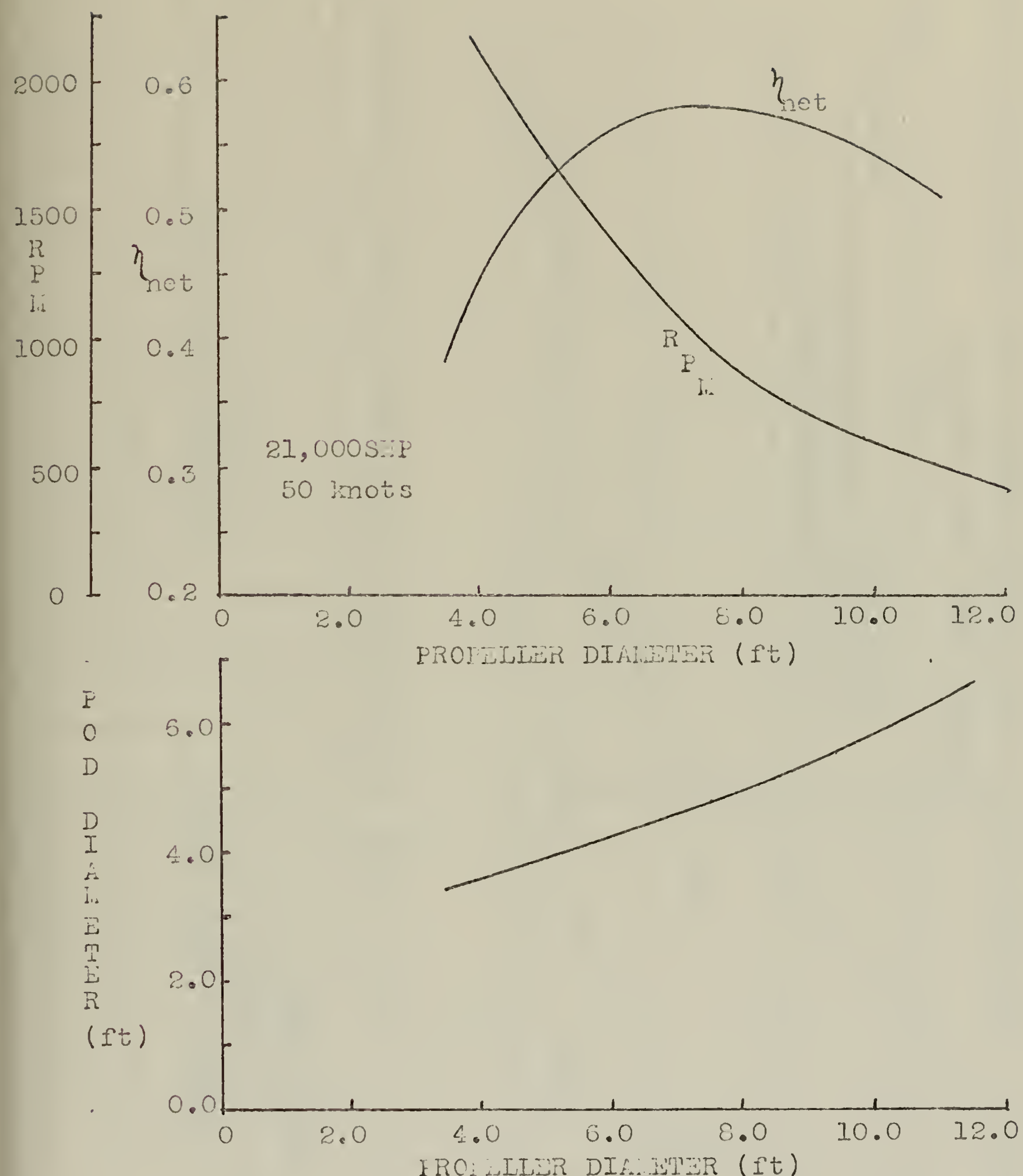


FIGURE 3. Performance of Supercavitating Propeller
(22) Pod Combination in Superconducting Sample

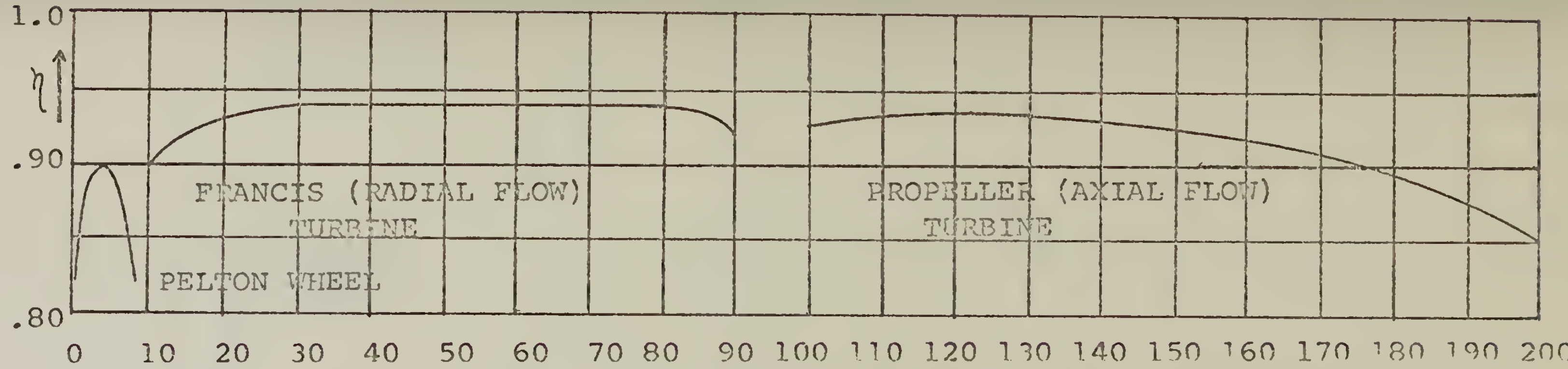


FIGURE 4a. SPECIFIC SPEED (CFS NOTATION) AND EFFICIENCY RANGES OF TURBINES (30)

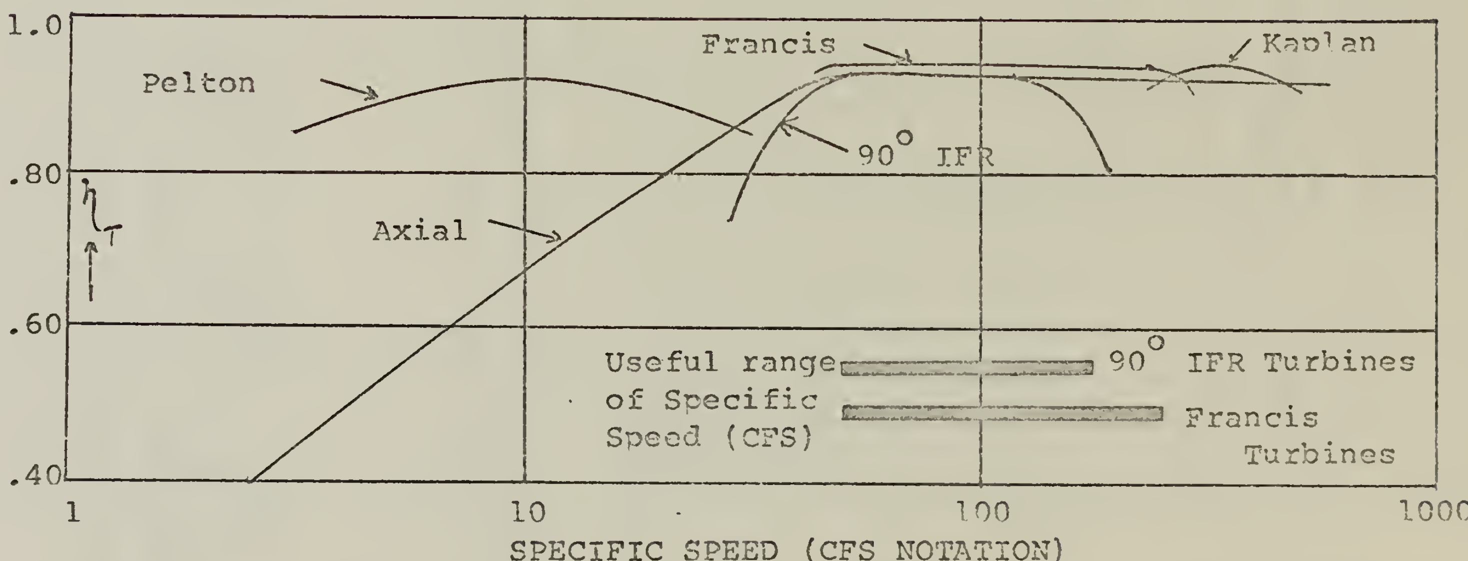


FIGURE 4b. COMPARISON OF 90° INFLOW RADIAL TURBINES WITH COMMON WATER TURBINES (28)

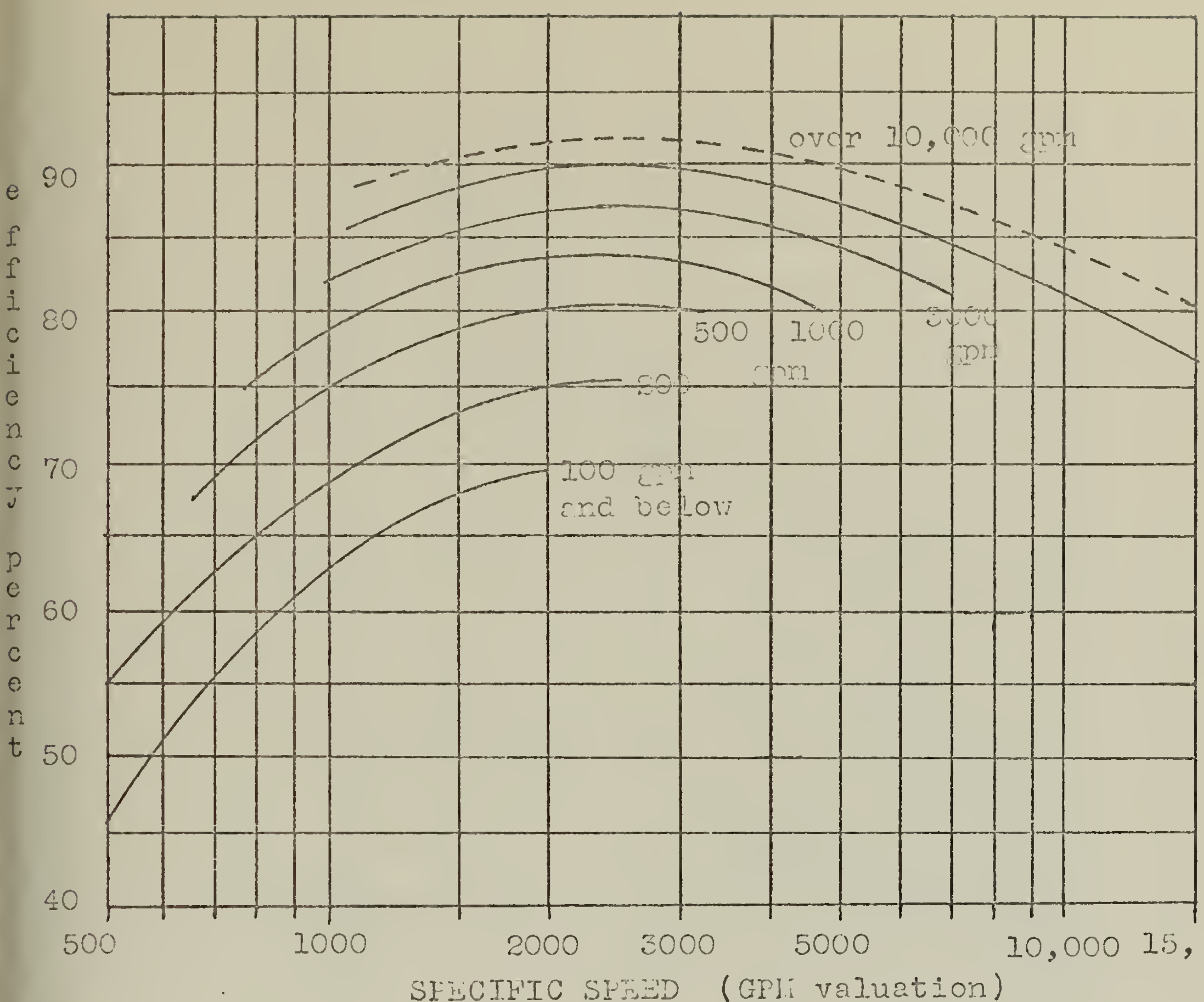


FIGURE 5. Variation of efficiency with specific speed
(30) for various sizes of pumps (Worthington Co.)

Form of Dynamic Pump Impellers Related to Specific Speed

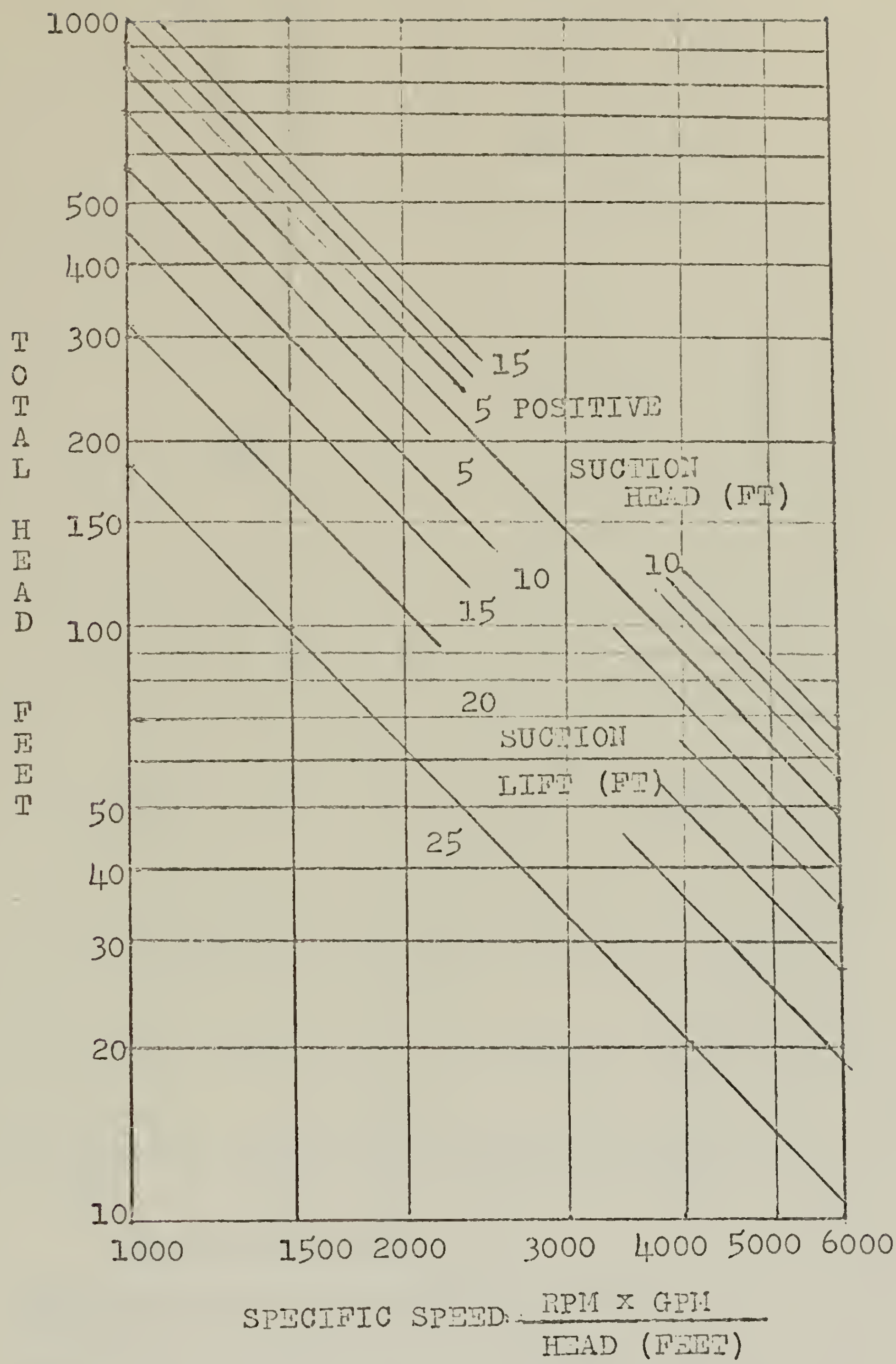


FIGURE 6. TOTAL HEAD VERSUS SUCTION
(29) HEAD AT VARIOUS SPEC. SPD.

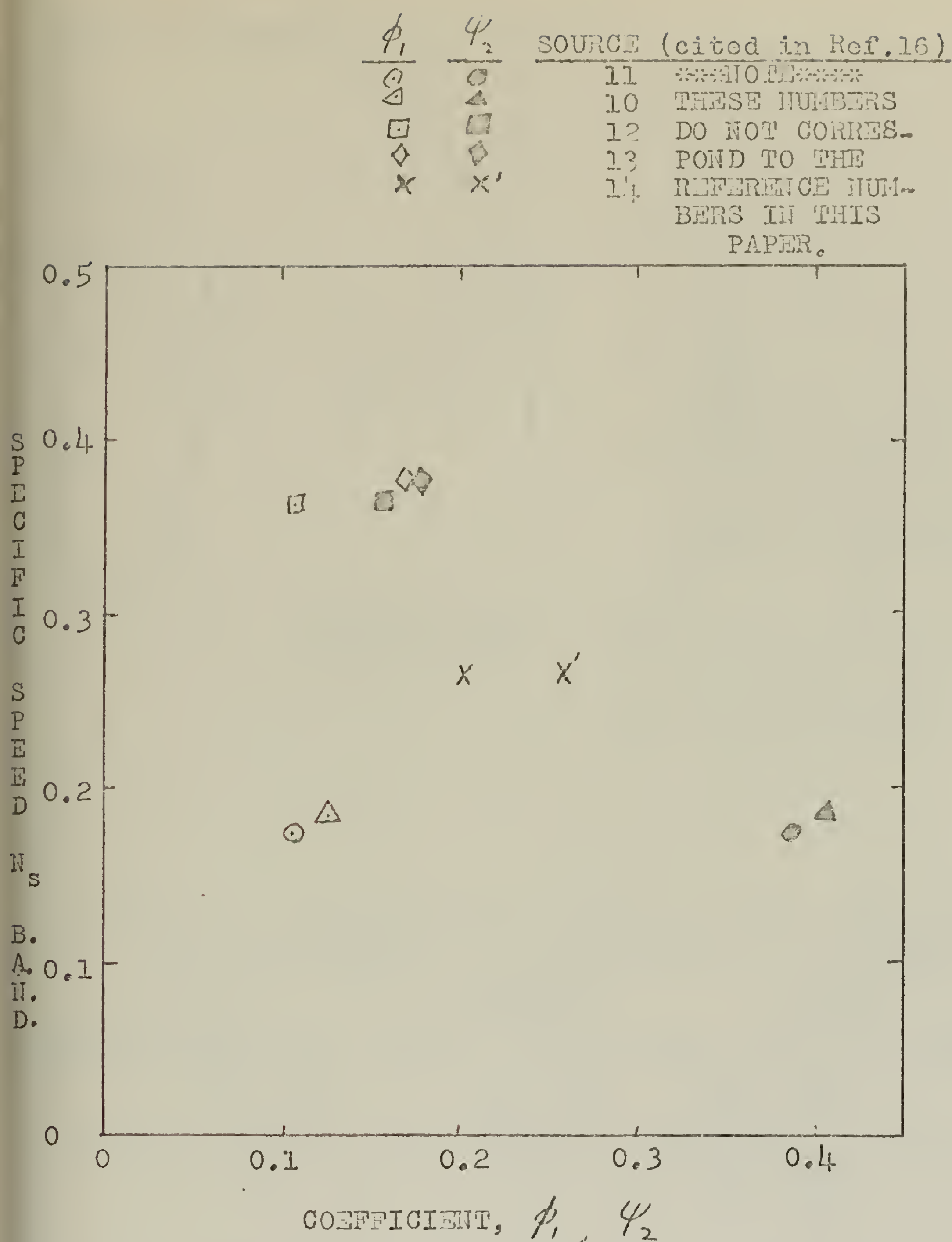


FIGURE 7. Inducer inlet flow coefficient and Head coefficient versus specific speed for various pump designs.

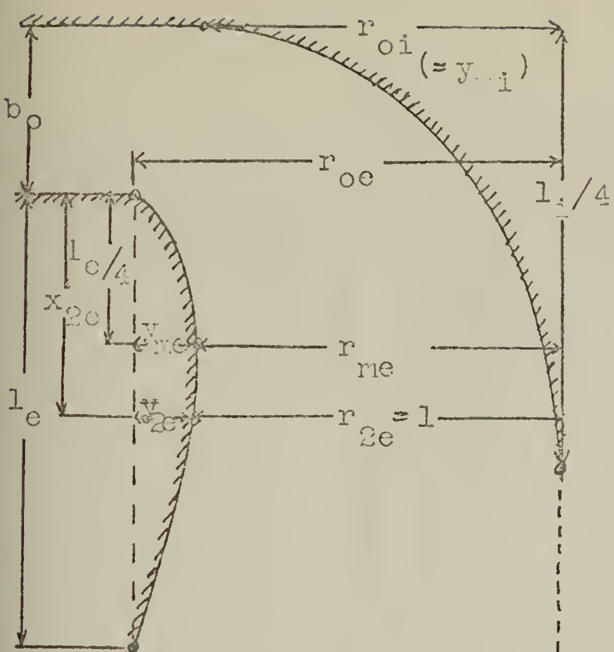


FIGURE 8a. RUNNER PASSAGE DIMENSIONS (21)

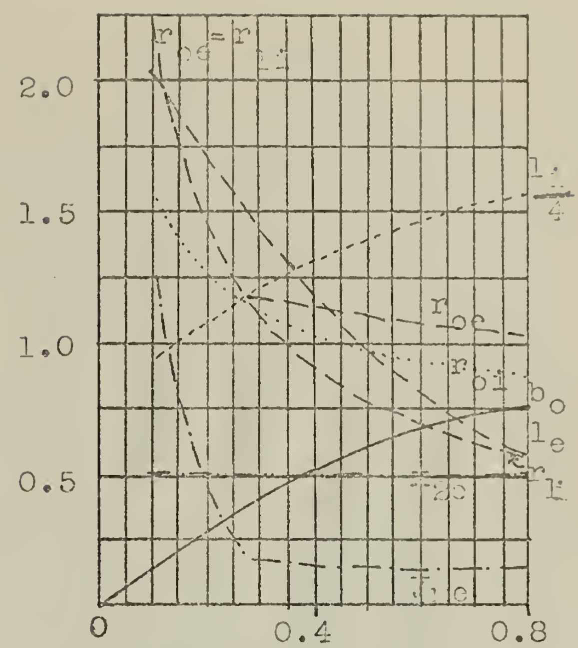
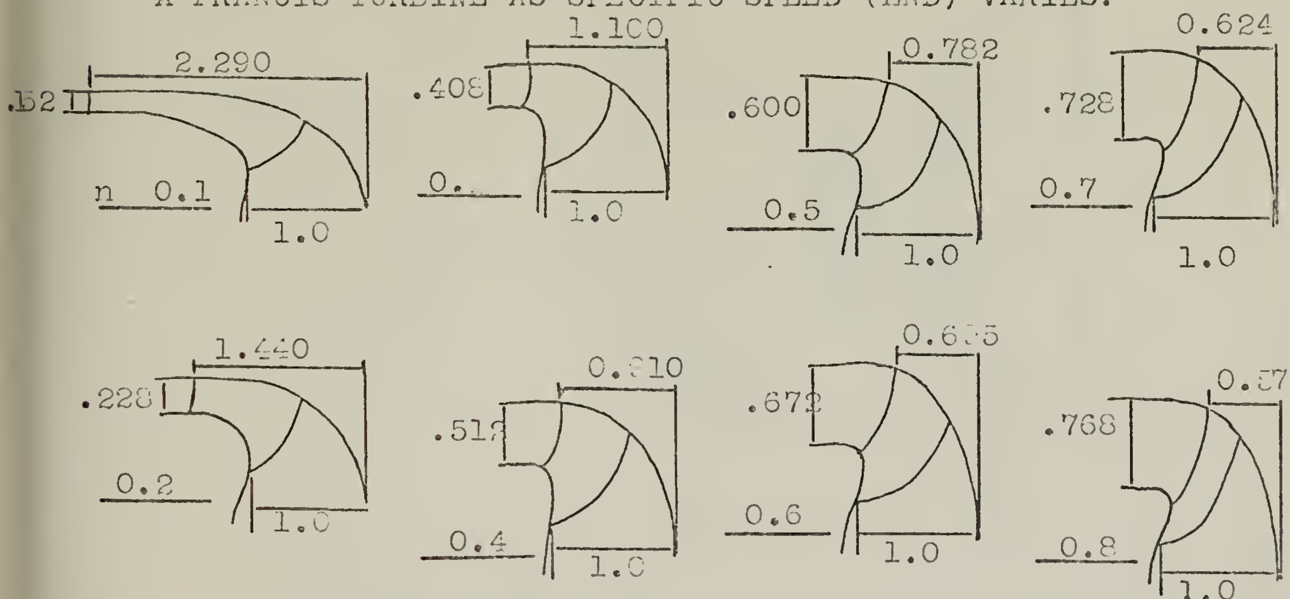
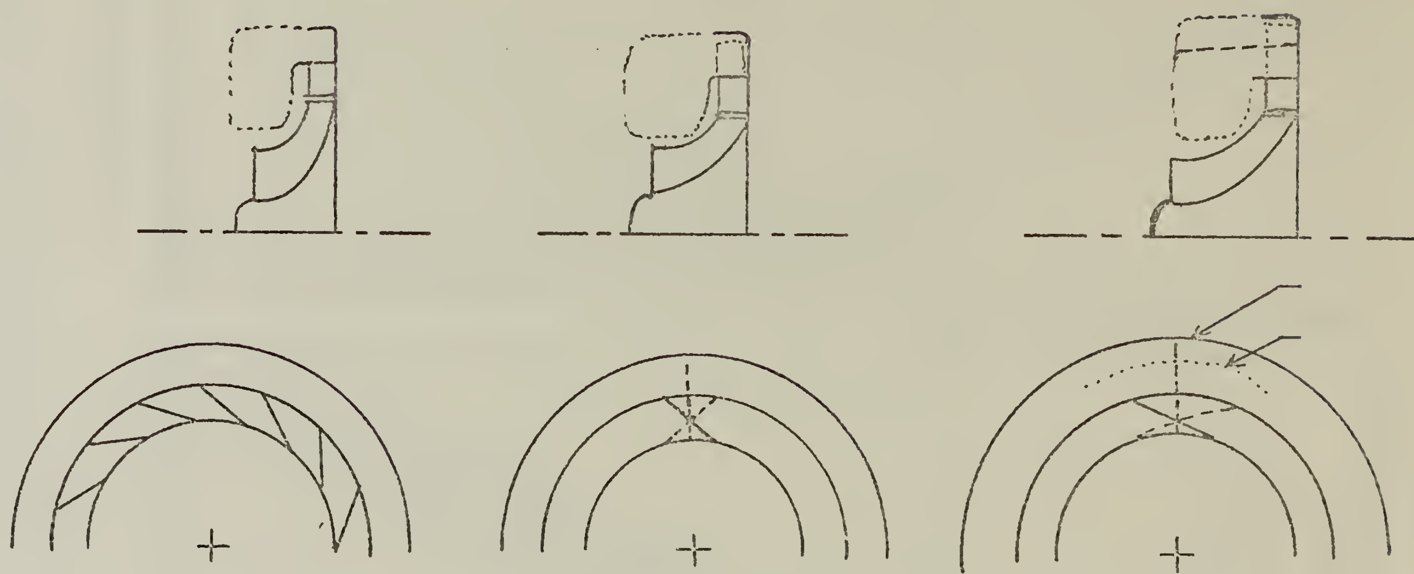


FIGURE 8b. VARIATION OF RUNNER PASSAGE DIMENSIONS WITH SPECIFIC SPEED (END)

FIGURE 8c. CHARACTERISTIC SHAPE OF RUNNER PASSAGE OF A FRANCIS TURBINE AS SPECIFIC SPEED (END) VARIES.





NON-REVERSING ARRANGEMENT (MINIMUM TOTAL CASING DIAMETER)

REVERSING 90° IFR ARRANGEMENT (TOTAL DIAMETER NOT INCREASED)

REVERSING FRANCIS ARRANGEMENT (INCREASED CASING DIAMETER REQUIRED)

(Possibility only. This consideration may or may not apply to a particular case.)

FIGURE 8d. EFFECT OF NOZZLE ANGLE ON DIAMETER OF REVERSING TURBINES

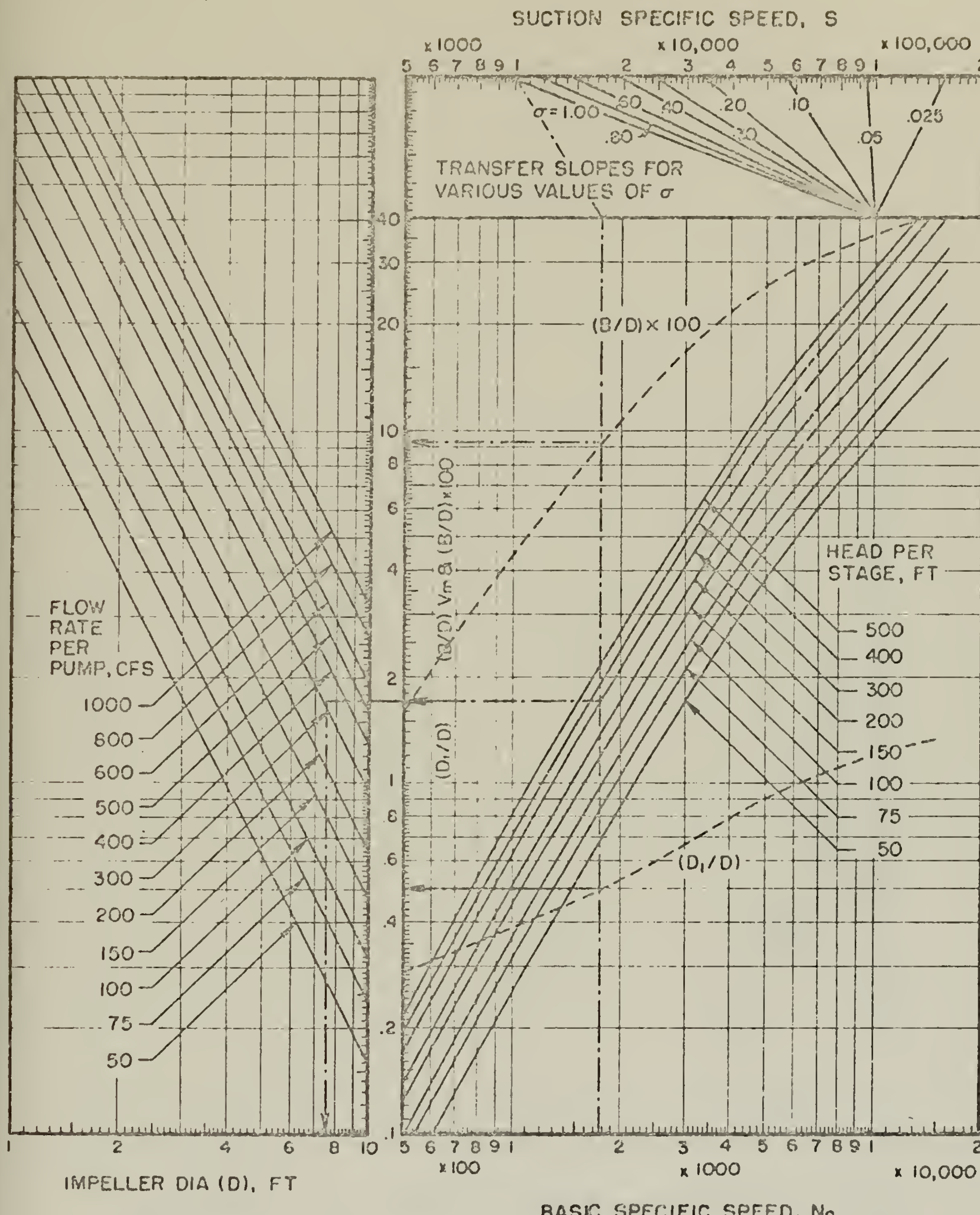


FIGURE 9. PUMP SIZING CHART FOR CONVENTIONAL PUMPS (35)

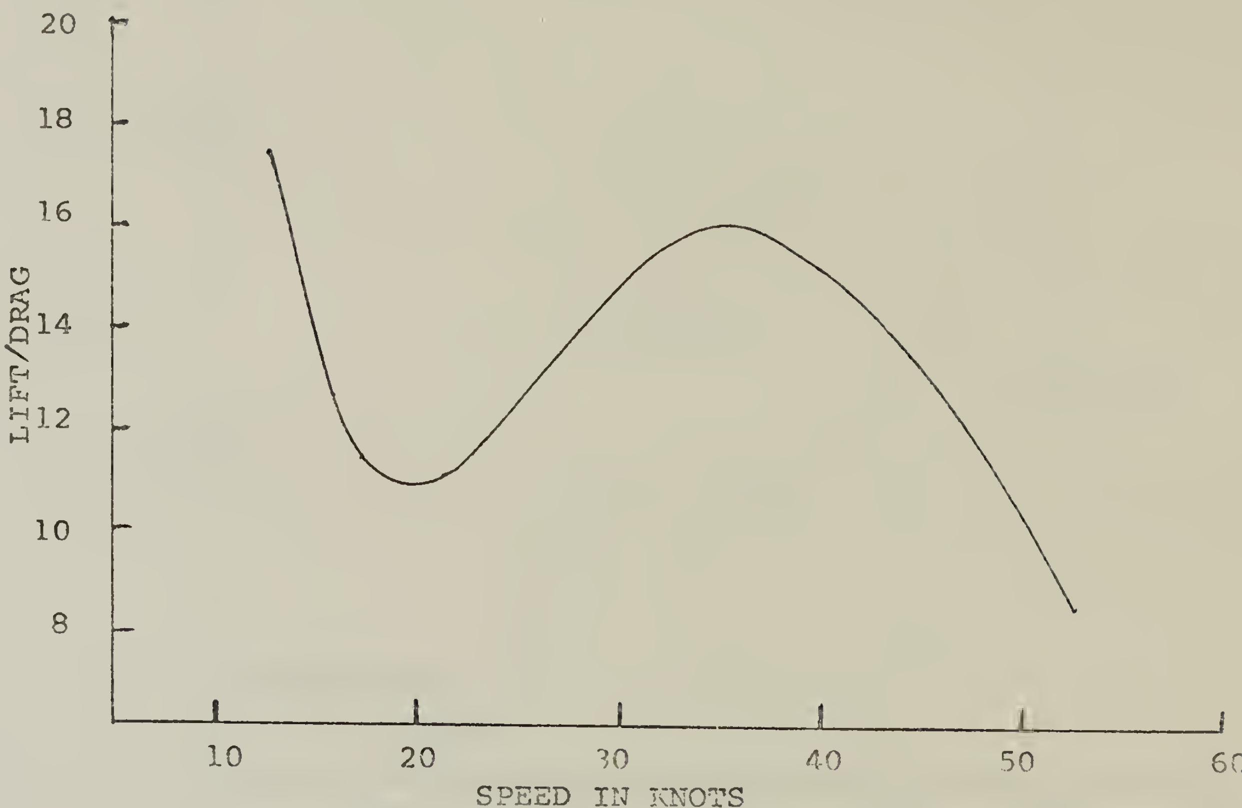


FIGURE 10a. TYPICAL HYDROFOIL VESSEL SPEED-POWER CURVE (22)

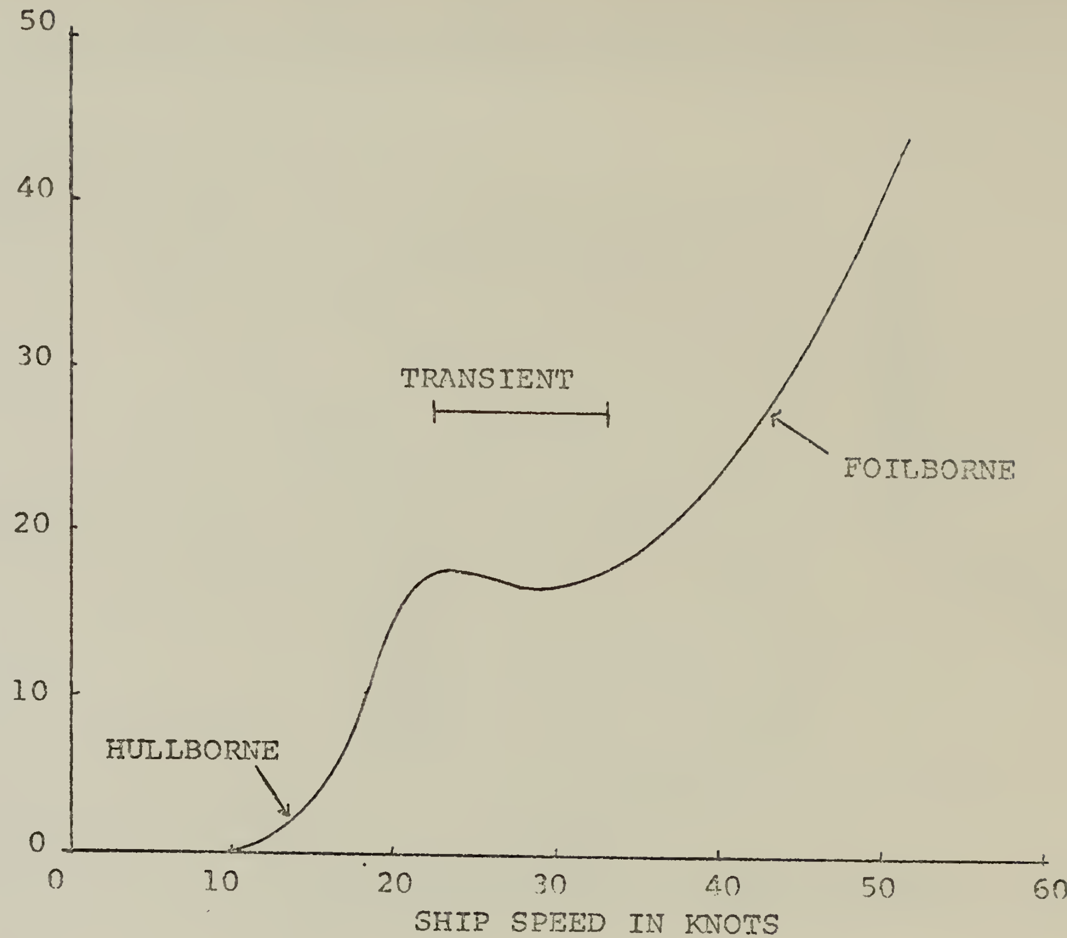


FIGURE 10b. DBH (SUPERCONDUCTING) SPEED-POWER CURVE

(22)

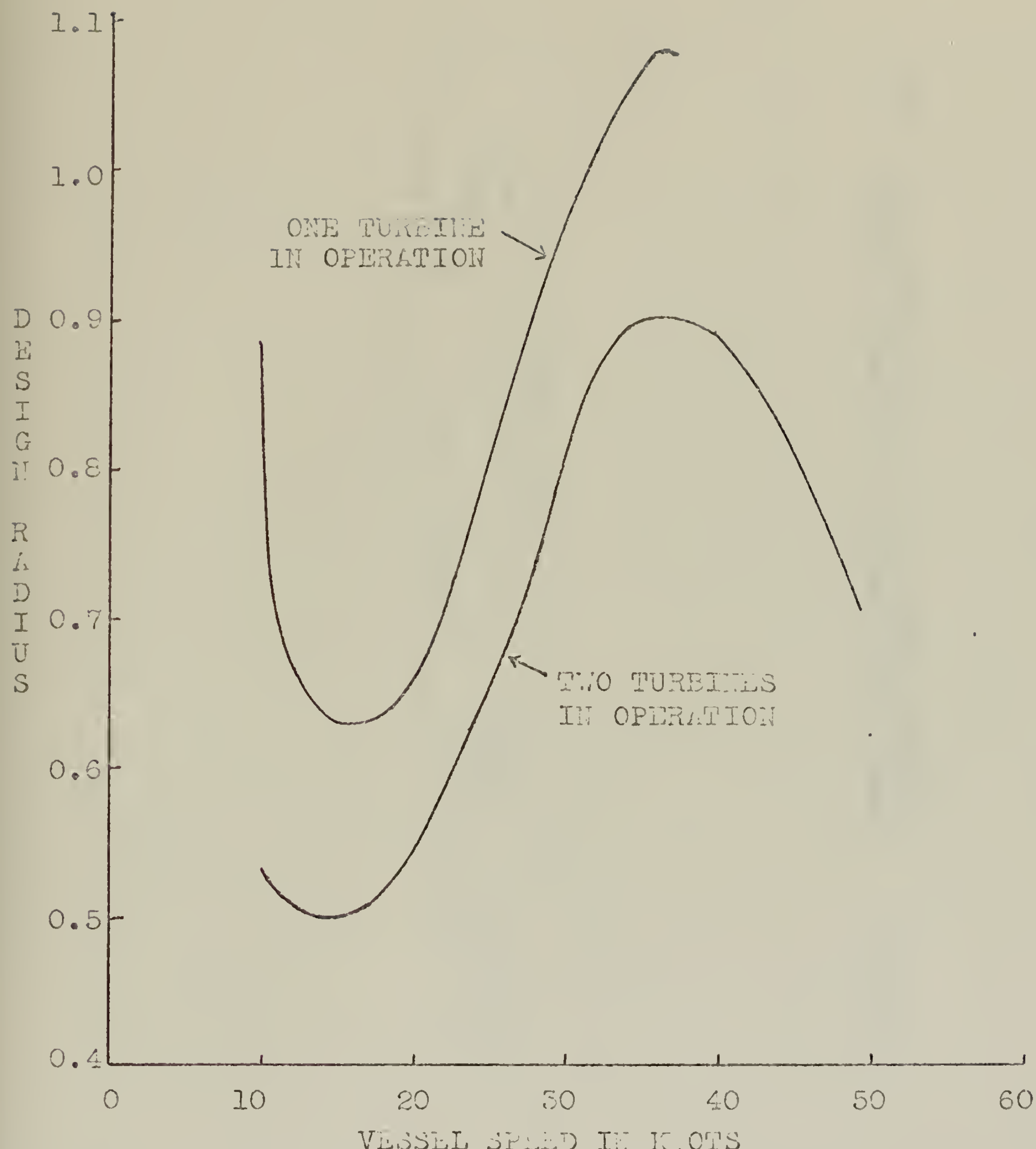


FIGURE 11. 750 TON HYDROFOIL CRUISING RANGE (SUPERCONDUCTING) (22)

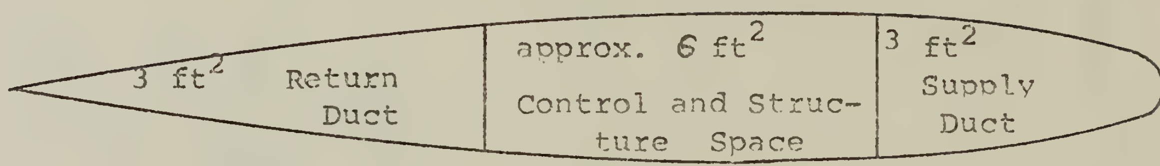
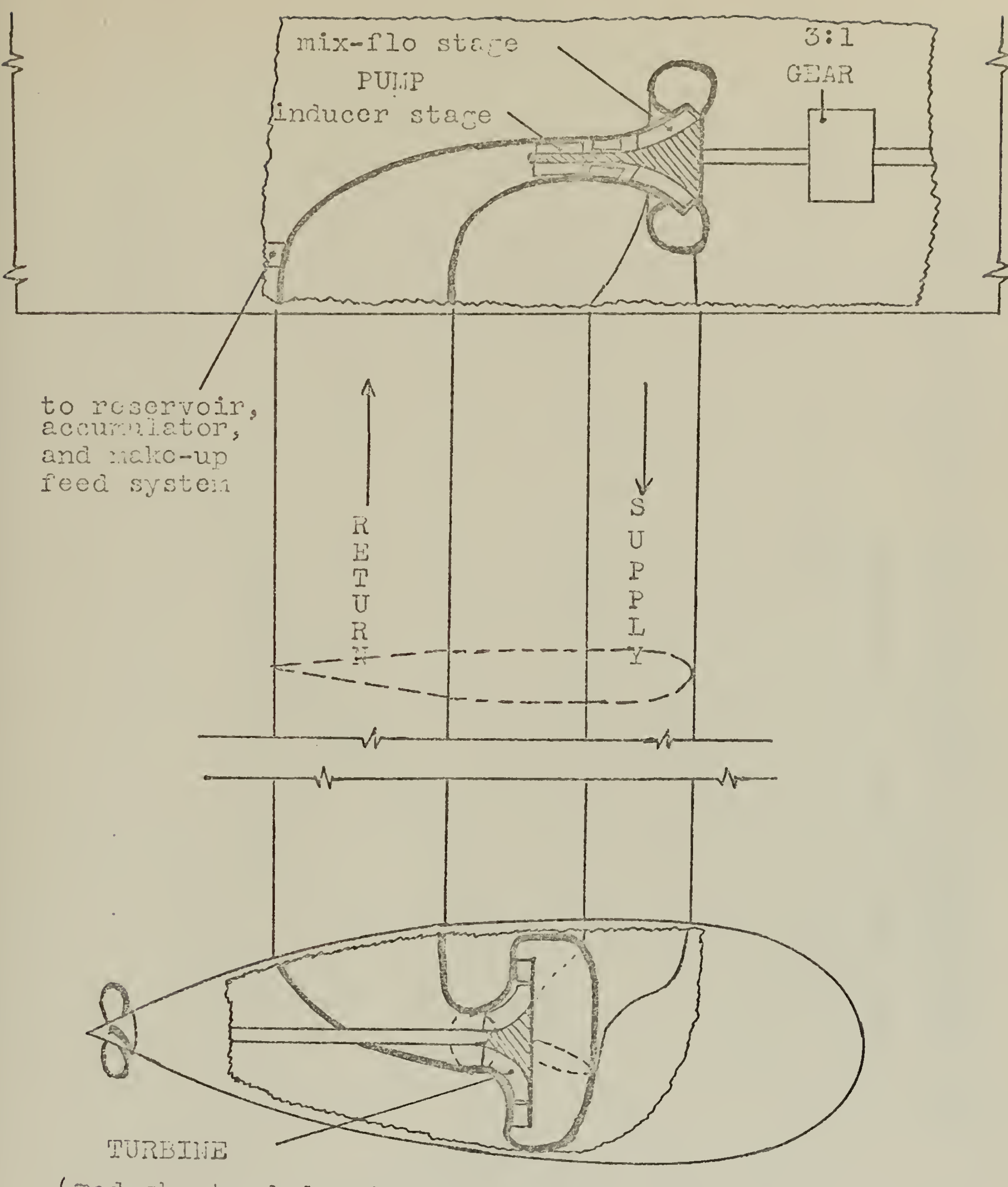


FIGURE 12 STRUT CONFIGURATION FOR SAMPLE VESSEL
(Basic dimensions of strut from Gill (15))



(pod shortended, widened. Turbine casing enlarged, for clarity. Propeller reduced.)
(Turning vanes and de-swirl vanes not shown.)

FIGURE 13. PROPOSED ARRANGEMENT OF HYDROKINETIC TRANSMISSION ON SAMPLE VESSEL (DBH)

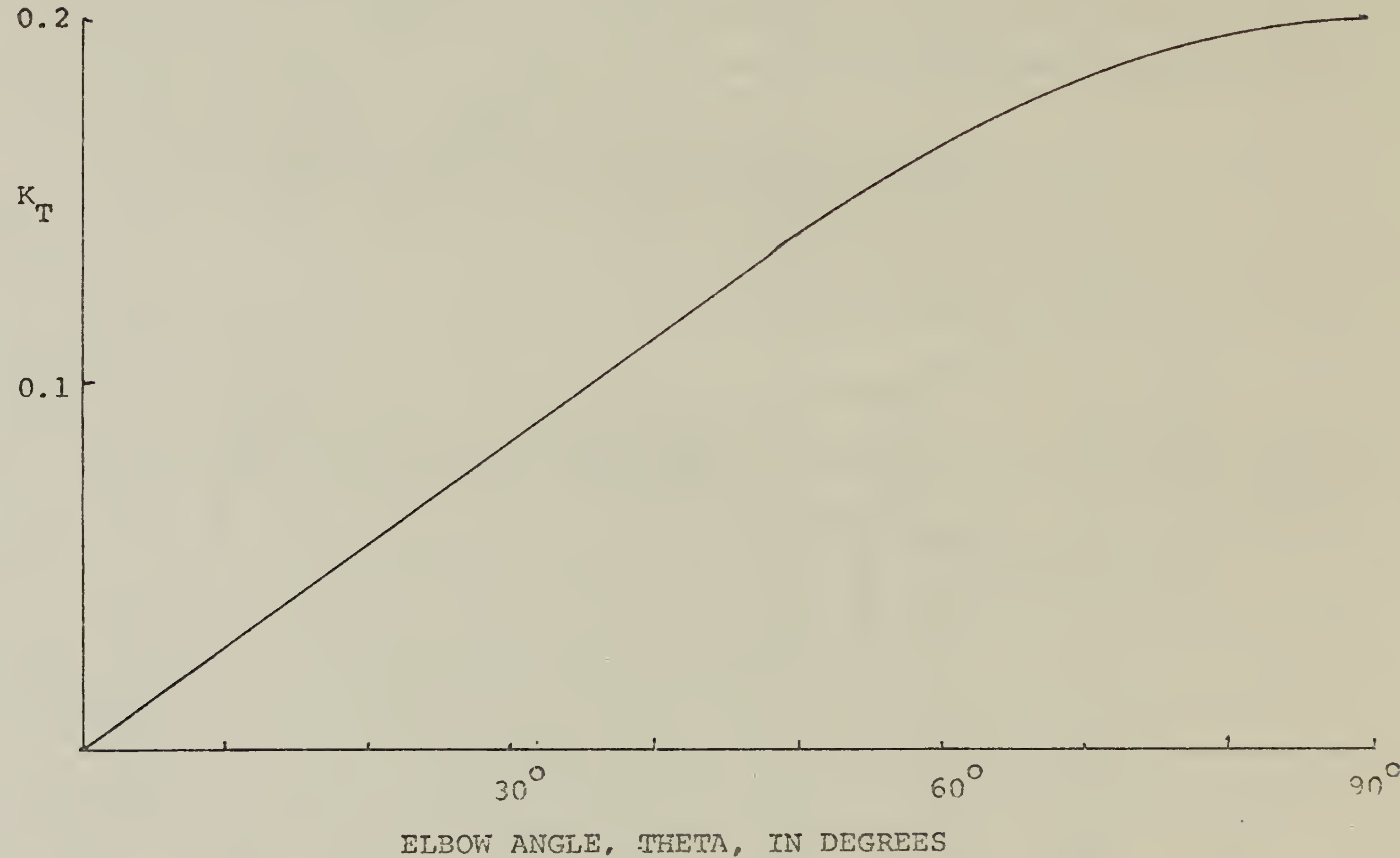


FIGURE 14a. LOSS COEFFICIENTS IN ELBOWS WITH THIN CIRCULAR-ARC VANES (15)

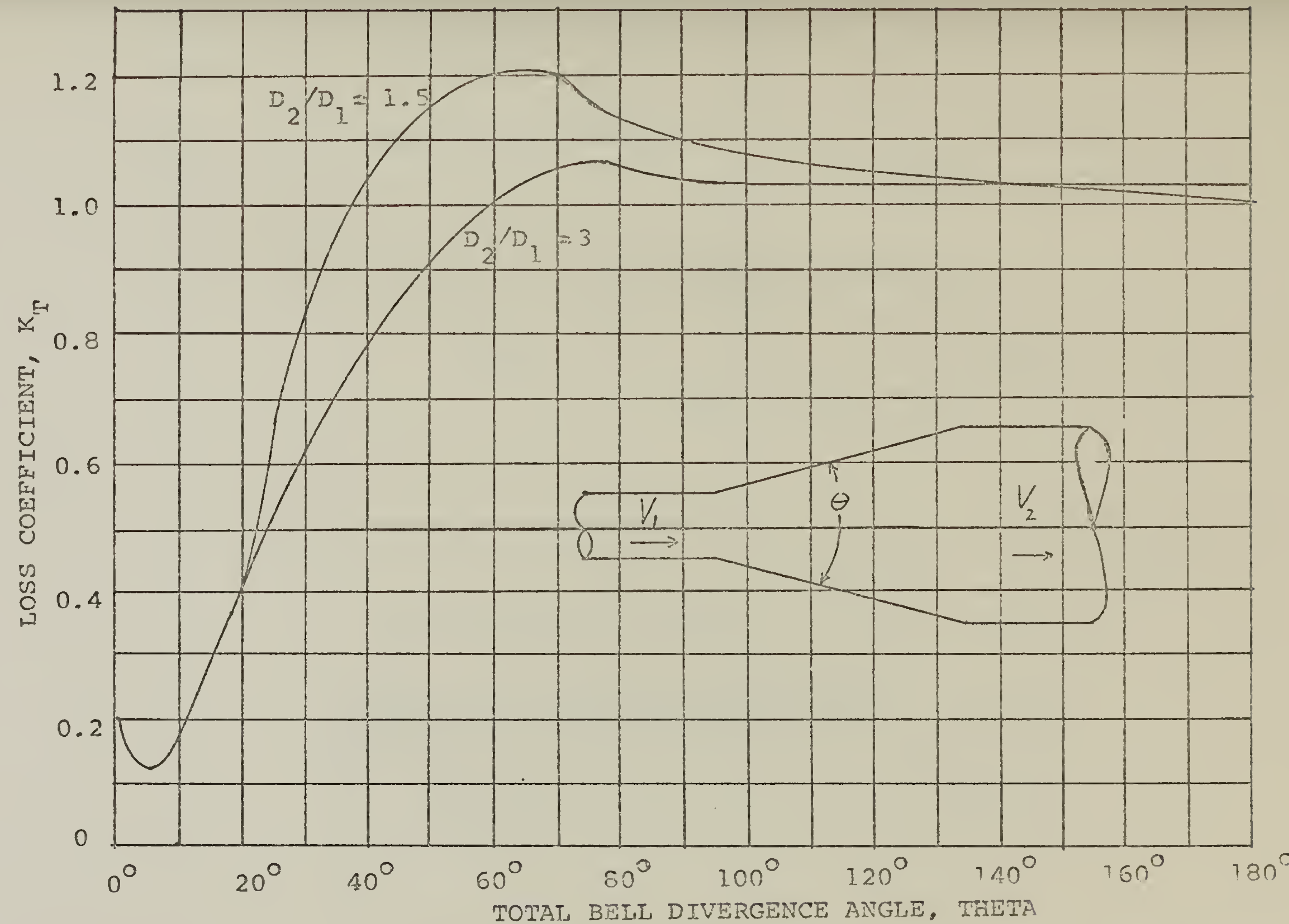
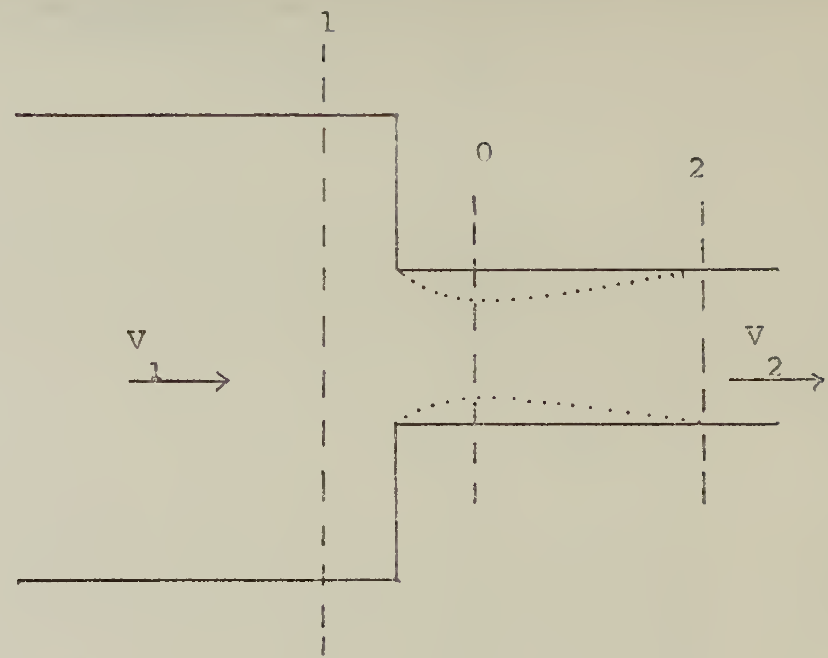


FIGURE 14b. LOSS COEFFICIENT FOR CONICAL EXPANSIONS OF VARYING AREA RATIOS (17)

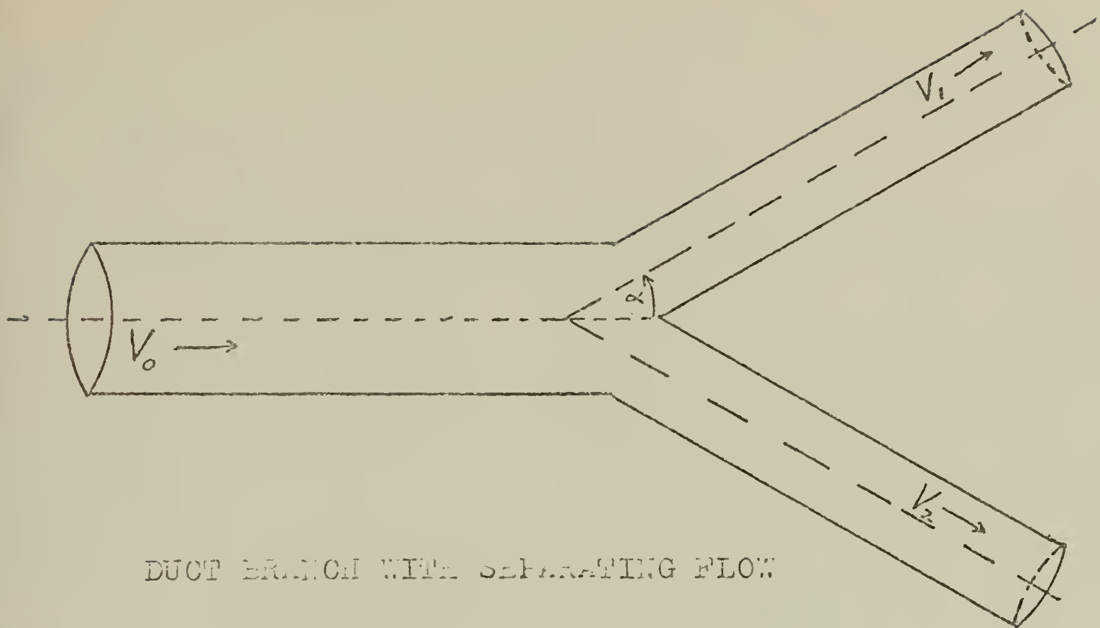


$$h_c = ((1/C_c)-1)^2 \; V_2^2/2g$$

CONTRACTION COEFFICIENT FOR WATER

A_2/A_1 ;	0.1	0.2	0.3	0.4	0.5	0.6	0.7	0.8	0.9	1.0
C_c ;	.62	.63	.64	.66	.68	.71	.76	.81	.89	1.0

FIGURE 14c. CONTRACTION COEFFICIENTS FOR SUDDEN CONTRACTIONS IN PIPES
(17)



DUCT BRANCH WITH SEPARATING FLOW

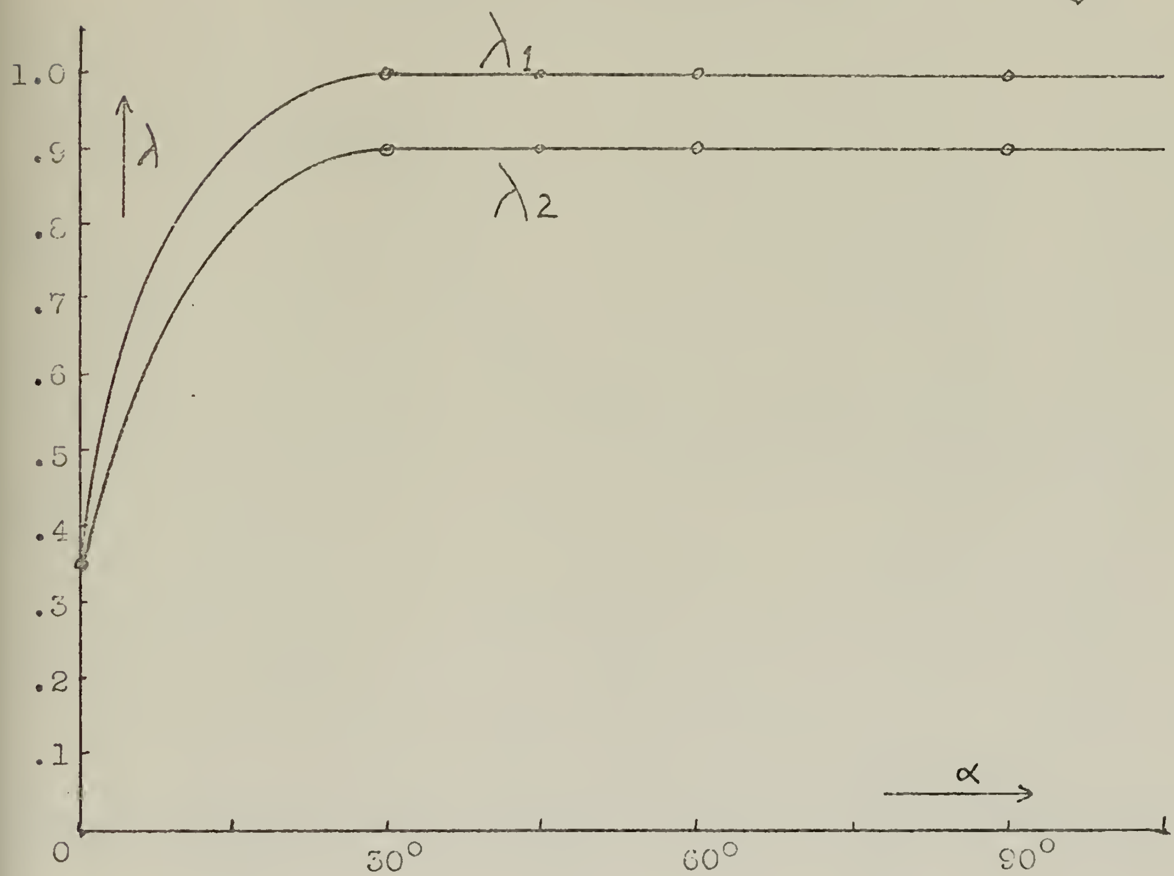
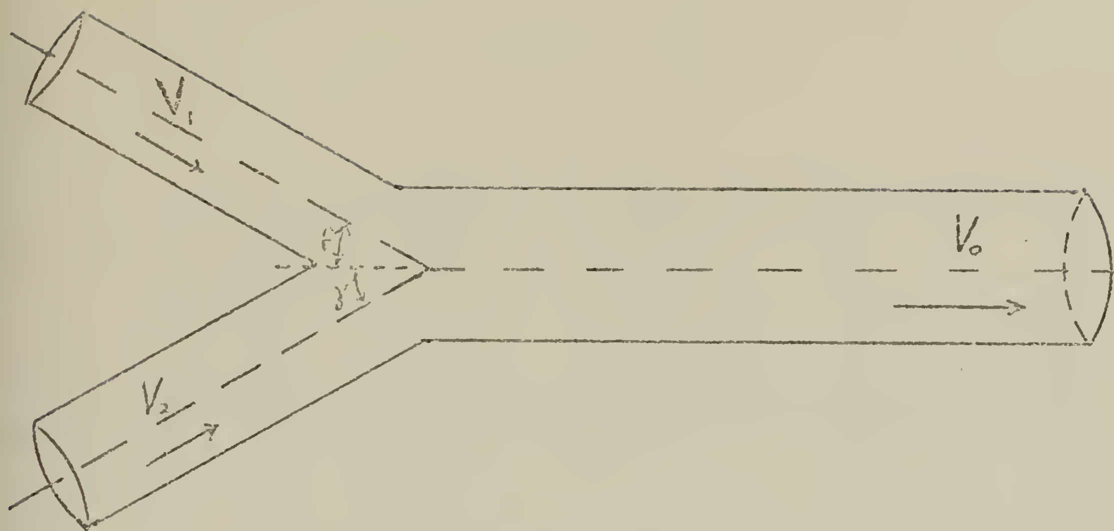


FIGURE 14d. LOSS FACTORS IN DUCT BRANCH WITH SEPARATING FLOW
(39)



DUCT BRANCH WITH UNITING FLOW

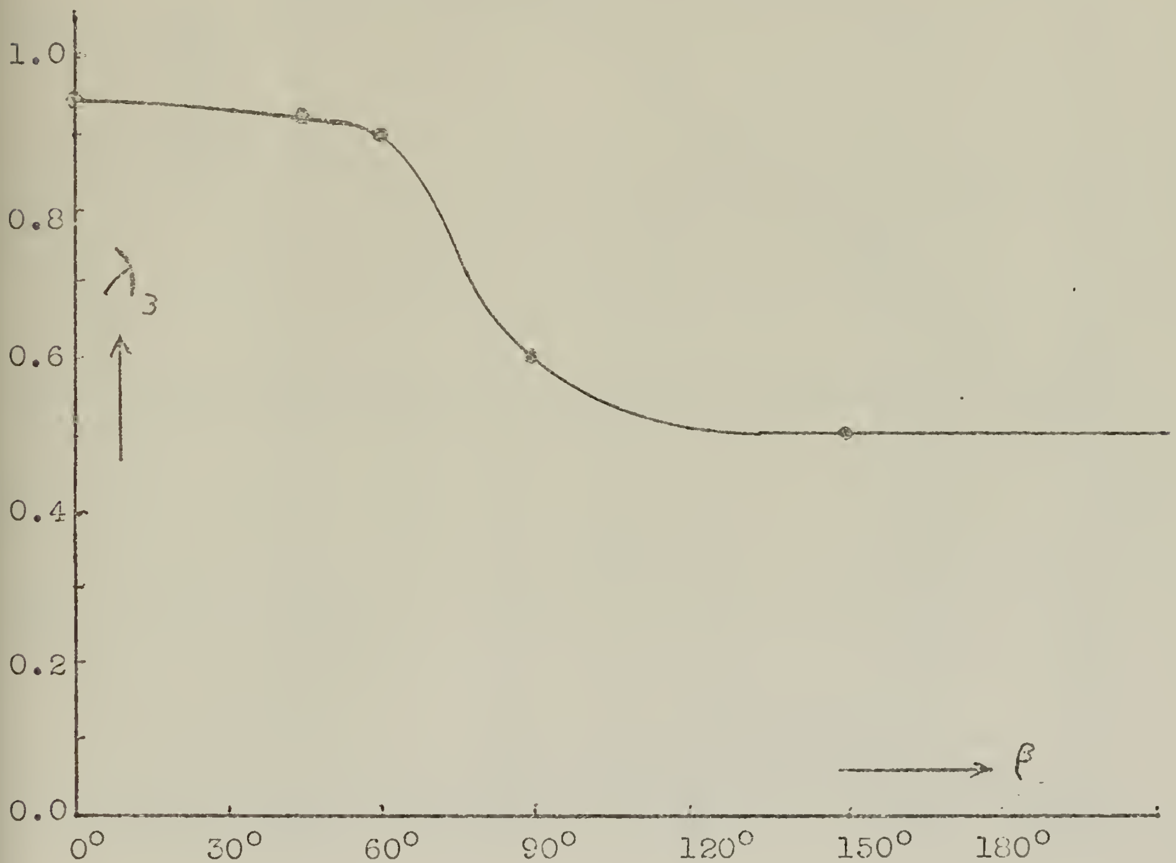


FIGURE 14e. PRESSURE LOSS IN DUCT BRANCH WITH UNITING FLOW (39)

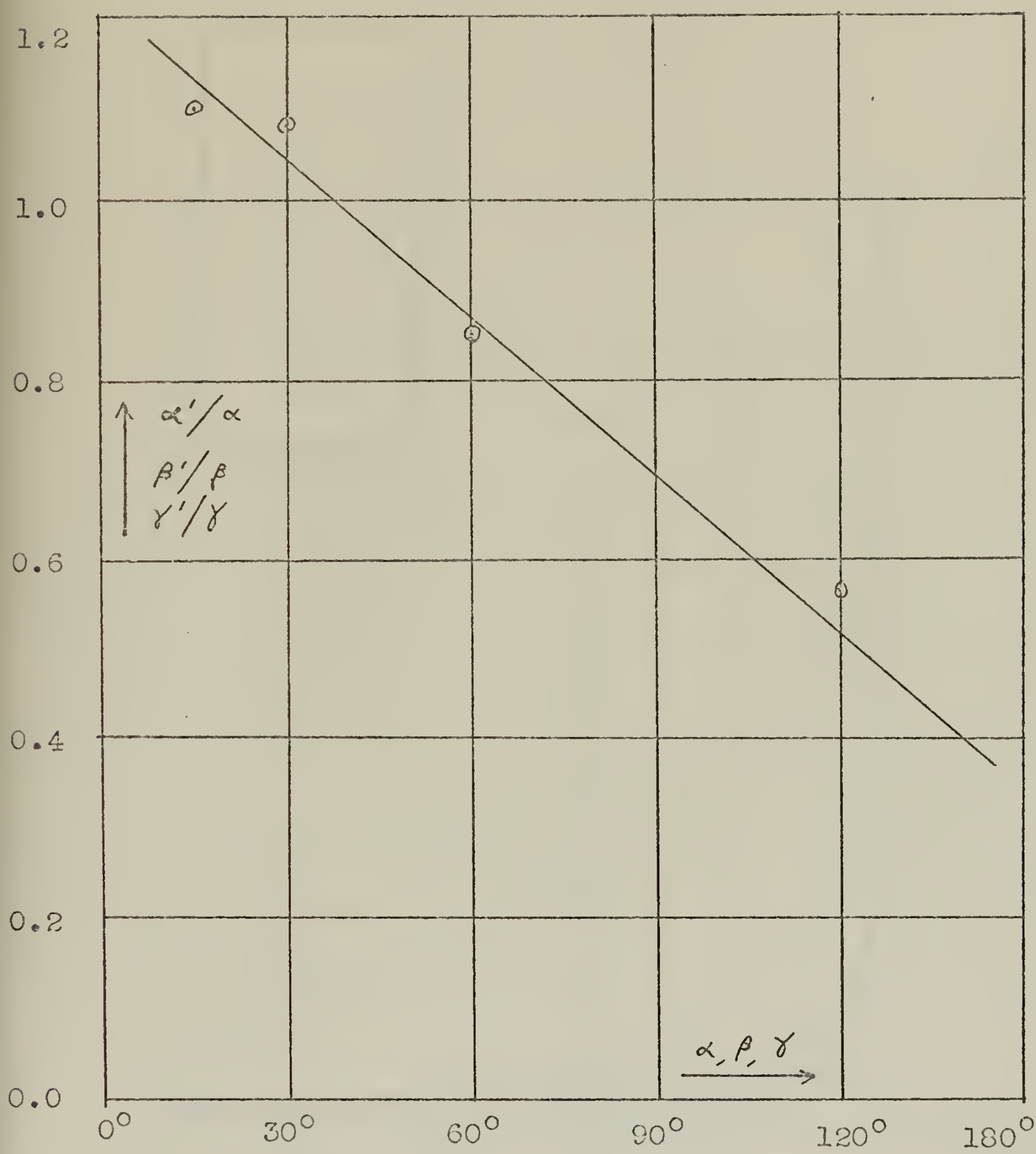


FIGURE 14f. EFFECTIVE ANGLE OF DEFLECTION AS A FUNCTION OF TRUE ANGLE OF DEFLECTION (39)

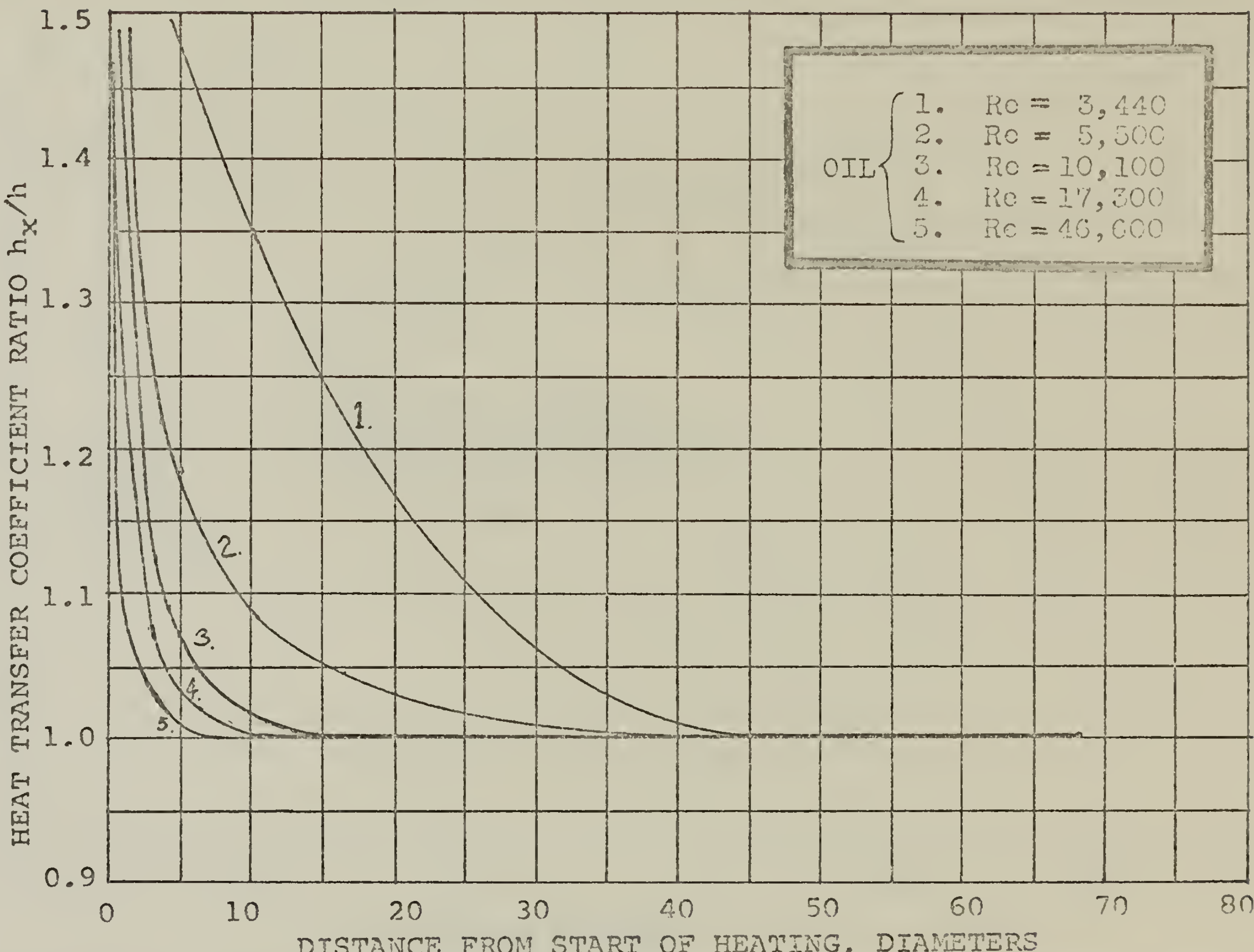


FIGURE 15. ENTRANCE EFFECT ON HEAT TRANSFER COEFFICIENT (19, 40)

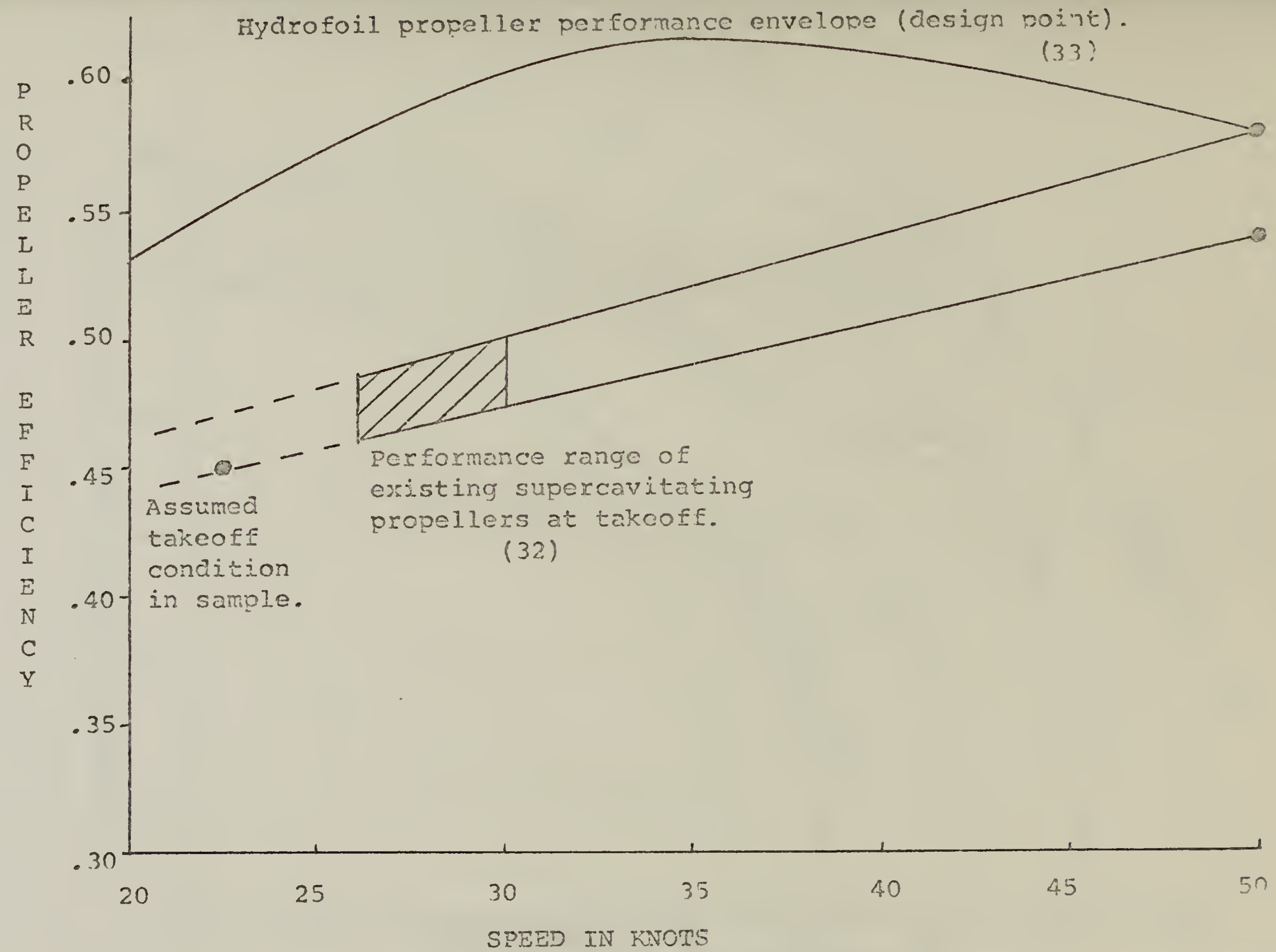


FIGURE 16a. ASSUMED PROPELLER PERFORMANCE MAP FOR SUPERCAVITATING PROPELLERS

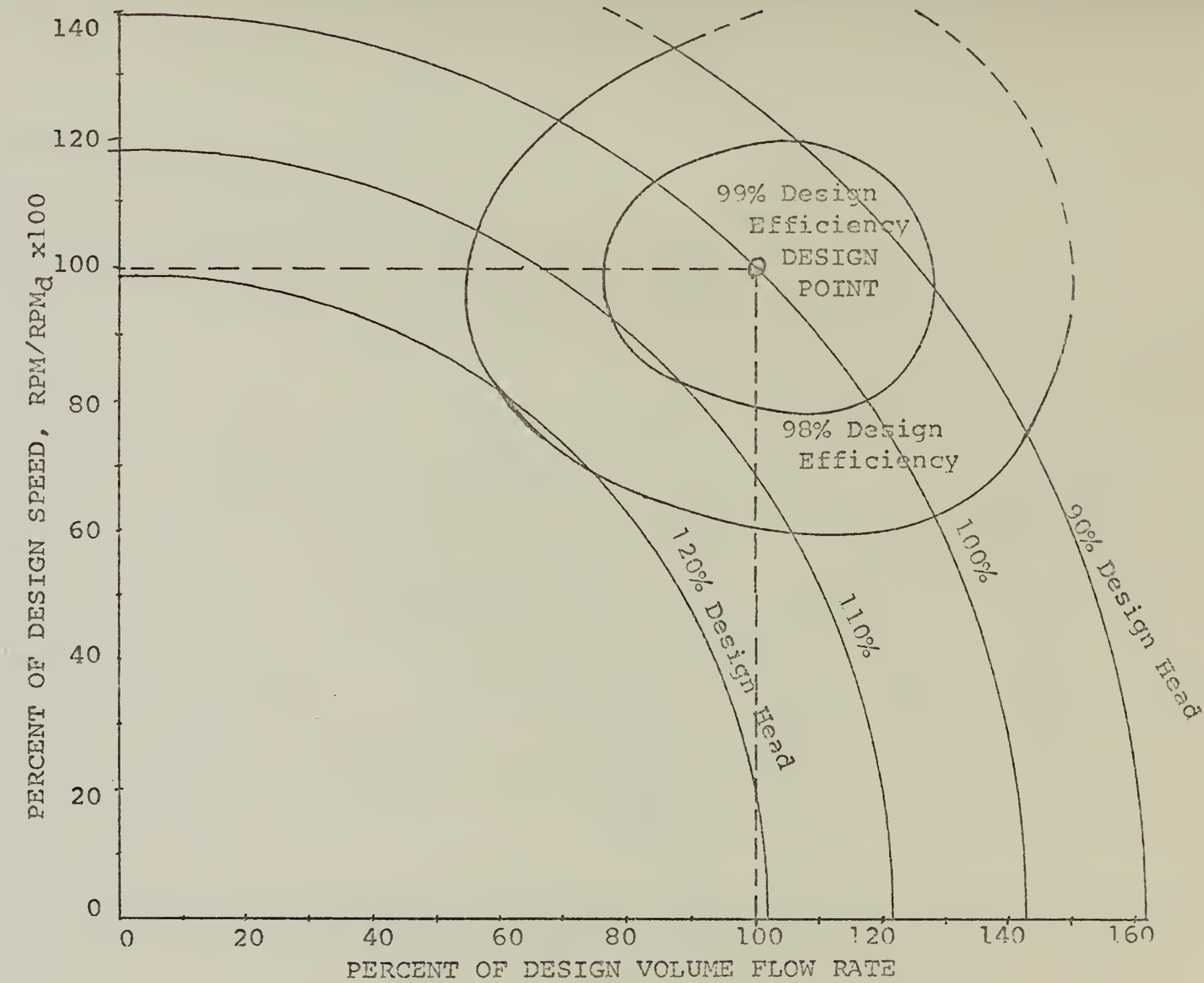


FIGURE 16b. ASSUMED PERFORMANCE MAP FOR RADIAL TURBINES

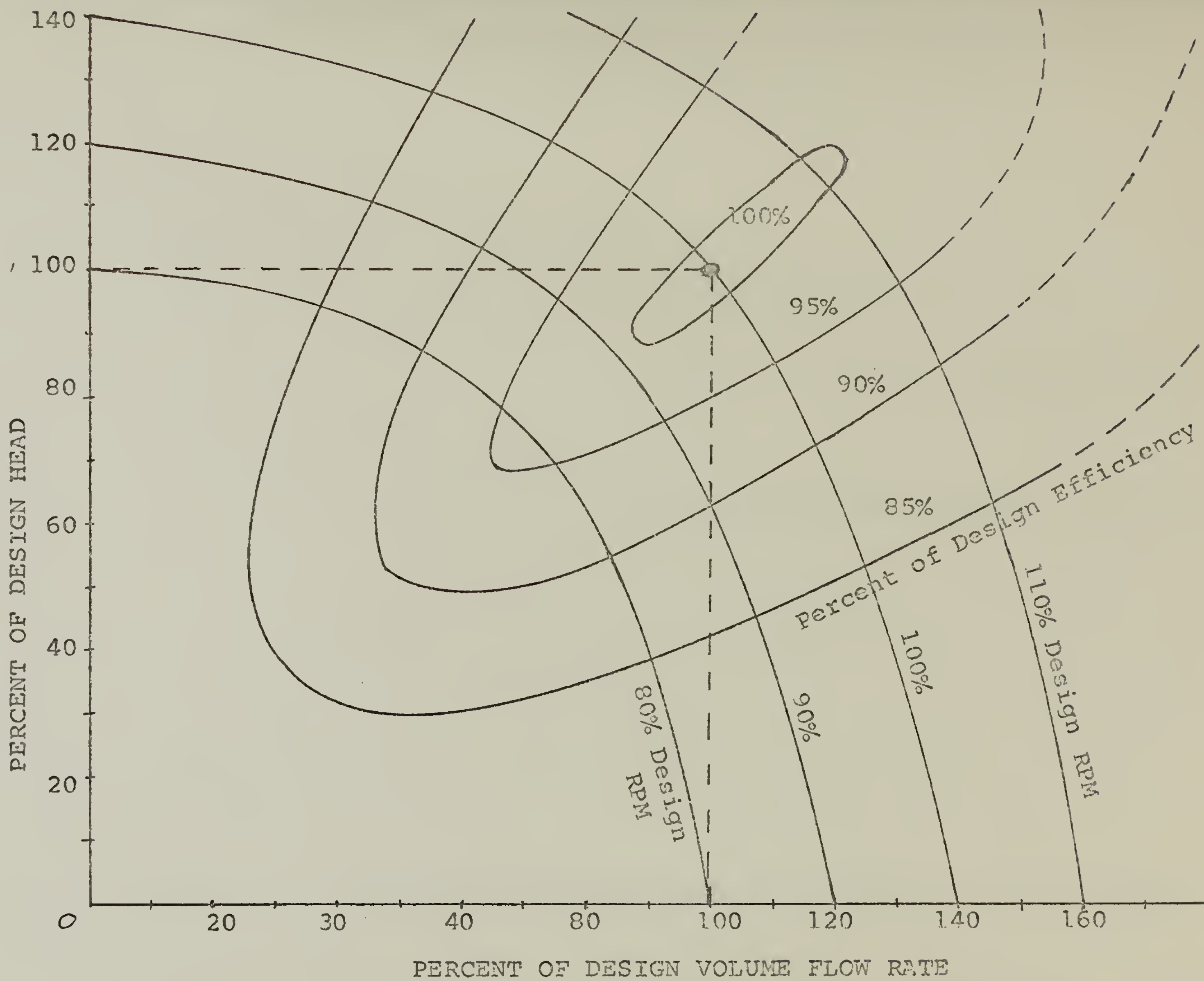


FIGURE 16c. ASSUMED PUMP PERFORMANCE MAP (SPECIFIC SPEED REGION 150(CFS))

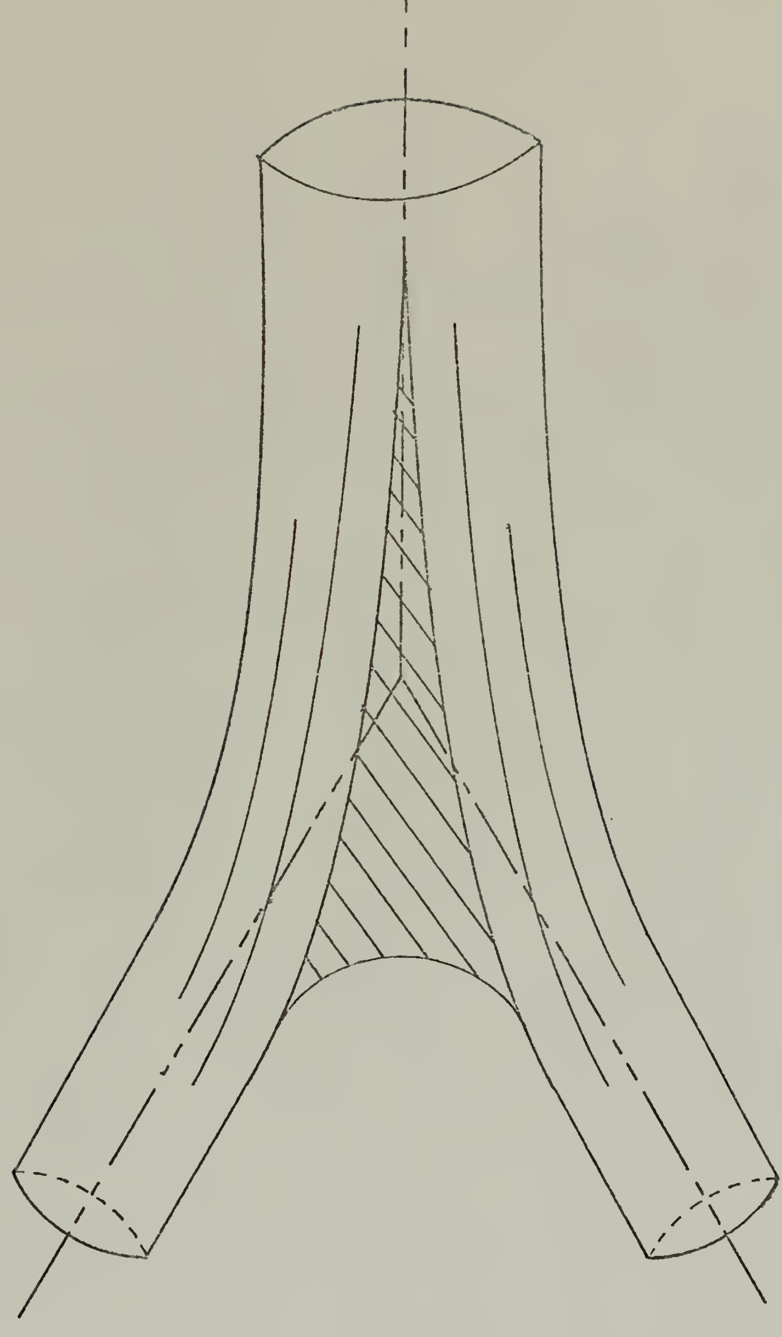


FIGURE 17. PROPOSED FORM OF BRANCHED CONNECTION TO REDUCE LOSSES

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J458 Jenkins
Hydrokinetic propulsion drives; a feasibility study.